A N   I N V E S T I G A T I O N

O F   T H E

H Y D R A U L I C   J E T   P U M P

A thesis submitted in partial satisfaction
of the requirements for the degree of
MASTER OF SCIENCE
at the University of California

by

HOWARD HAMILTON BLISS

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Object - Description of Pump

It was the purpose of the work herein described to study the operation of the hydraulic jet pump in the laboratory of the University of California. The investigation included finding the loss coefficients in each part of the pump, testing various arrangements of the parts for efficiency, and studying the rate at which tha parallel jets of water unite to acquire a common velocity. An attempt was made to test experimentally the accuracy of a theoretical expression for the impact losses, which are very large in this kind of apparatus. Finally, certain relations were analytically developed for the operation of the device, and tests and calculations made to check them practically.

This jet pump has two concentric cylindrical chambers, the inner one for the high pressure working water and the other for the low pressure water to be lifted. The liquid emerges from these compartments through concentric nozzles, the low pressure water coming through the annular orifice surrounding the other. Because of the difference in head, the jet in the center has a higher velocity, and the operation of the pump depends upon the communication of this to the surrounding water. The streams mingle in the mixing chamber and then enter the diffuser, where the kinetic energy is largely converted into pressure.

The pump tested has a number of mixing chambers of different lengths, any one of which can be set between the nozzles and the diffuser. They are all cylindrical, the diameter at every point being five-eighths inch, which is also the diameter of the outer nozzle and of the throat of the diffuser.

In making the tests described below, the low pressure water was taken from a storage tank by a centrifugal pump which delivered it to a stand pipe, whence it flowed through piping and valves to the jet pump. The rest came from the tank through a Quimby screw pump and a water meter. The delivery water was usually weighed and sent into the storage tank. For some of the tests, however, it was allowed to issue into the open air and was diverted downward into the weir box on which the jet pump was located. The other apparatus used consisted of two pressure gauges, a mercury manometer, a water manometer, a watch, and a Pitot tube built especially for this pump.

It will be necessary in this report to express by formulas the relations of various quantities measured and computed. After trying in vain to make satisfactory use of a system of nomenclature with numerical subscripts, I have developed a system using the letters s, e, r, j, m, t, d, and f as subscripts. A glance at sketch No. 1 will make clear their use. Thus Q_s signifies the cubic feet of suction water per second, v_s its velocity, and h_s its pressure head (measured in feet of water above an absolute vacuum). Subscript e indicates the high pressure water entering the pump, r signifies the annular nozzle stream, j the inner jet, m the mixing chamber, t the throat of the diffuser, d the diffuser, and f the final condition as the water leaves the apparatus. The letter a indicates area of cross section in square feet, and ζ is the loss coefficient.

EFFICIENCY TESTS

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On account of the high speed at which the water passes through the mixing chamber, it was expected that a considerable loss of energy would occur, due to friction there. Hence a variation

in the length of the chamber would change the efficiency, as this loss would increase with increasing length. However, since the entire operation of the pump depends upon the imparting of the kinetic energy of the working water to the surrounding stream, too short a chamber would also destroy the efficiency.

These assumptions were abundantly justified by two series of tests with varying lengths. In the first series the working water was always admitted to the pump at 70 lbs. per □" above atmosphere and the suction water entered at atmospheric pressure. In the second series the pressures were 39.4#/□" and atmospheric. With each mixing chamber twelve or more runs were made with different delivery pressures. For each run the data taken consisted of the delivery pressure, the weight of water delivered, the cubic feet of high pressure water used, and the duration of the run. All pressures were kept constant and verified several times during each test. The gauges were calibrated practically every day and four tests of the water meter at different times showed that its readings were reliable within the limit of errors of observation.

The efficiency, η, is computed as foot lbs. of work done on the water lifted ÷ foot lbs. of energy lost by the working fluid. Letting $\underline{t}$ represent the seconds, $\eta = \frac{t\ Q_s(h_f - h_s)}{t\ Q_e(h_e - h_f)} = \frac{(t\ Q_f - tQ_e)p_f}{(t\ Q_e)(p_e - p_f)}$, where $\underline{p}$ indicates gauge pressures, p_s being 0.

On curve sheets Nos. 6 to 10 inclusive will be found plotted the results of these tests upon chambers varying in length from 0 to 6 inches. Under otherwise constant conditions the efficiency varies with varying delivery pressure, reaching a maximum when this is about 34 % of the high pressure (by gauge). The

greatest efficiency with zero mixing chamber is 22.7 %. Each succeeding longer chamber shows better operation until the length becomes 1 3/8 inches. For this and those of lengths 2", 2 1/2" and 3" the maxima are almost equal within the limit of probable error at about 31 %. Beyond three inches the efficiency falls gradually until it reaches 25 % for the six inch chamber. This is shown graphically on curve sheet No.10 , which is plotted from the previous curves.

Tabulated below will be found the data of these runs. In most cases more runs were made than recorded here, but the more inefficient ones were omitted. It will be noticed that the time is recorded, though not used in computing efficiency. It was measured necessarily while making the runs - as most of the time I had no assistant and had to read the water meter a definite number of seconds after attending to the weighing - and is inserted here with the idea that it may be useful for later study from this data. The series on the two inch chamber was run last and most carefully, the time being extended more than ordinarily to decrease the effect of errors in reading the meter.

Mixing chamber 0" long.

$p_e = 39.4\#/\square''$					$p_e = 70.0\#/\square''$				
p_f	$t\,Q_e$	$t\,Q_s$	η	t	p_f	$t\,Q_e$	$t\,Q_s$	η	t
17.5	7.2	0.8	2.9	131	17.3	9.9	6.1	20.2	126
15.7	6.7	1.3	12.9	120	19.2	9.0	5.4	22.7	116
14.1	6.3	1.7	15.0	111	21.2	7.8	3.4	18.9	100
12.4	5.7	2.3	18.6	99	23.3	6.0	2.0	16.7	78
10.8	7.0	4.2	21.8	120	25.4	8.9	2.3	14.7	118
8.9	5.5	4.1	21.8	89	28.3	6.8	1.2	12.0	92

Efficiency tests tabulated

Mixing chamber 1" long.

p_e = 39.4 #/□"					p_e = 70.0 #/□"				
p_f	tQ_e	tQ_s	η	t	p_f	tQ_e	tQ_s	η	t
					19.7	7.1	4.1	22.6	92
					23.4	7.6	3.6	23.8	100
					25.2	5.6	2.4	24.2	75
					27.1	9.2	3.6	24.7	124
					28.9	9.7	3.1	23.5	131
					31.1	9.3	1.9	16.4	124

Mixing chamber 1 3/8 " long

18.8	10.5	2.3	20.0	190	21.3	7.0	4.2	26.2	95
17.5	6.1	1.9	24.9	108	23.3	12.1	7.2	29.7	155
15.8	5.6	2.4	28.7	98	25.4	7.4	3.8	29.2	97
13.5	7.9	4.9	32.4	138	27.5	5.3	2.7	33.0	73
11.6	6.7	4.5	28.0	112	29.5	11.5	4.5	28.5	153
10.0	5.7	3.9	23.2	90	31.5	8.4	2.8	27.3	114

Mixing chamber 2" long

9.3	9.8	7.8	24.7	174	32.8	18.3	2.5	12.0	273
12.2	9.7	6.3	29.1	178	30.4	15.0	4.2	21.5	223
15.0	10.65	5.35	30.9	203	27.1	11.1	4.9	27.9	163
17.5	13.9	3.7	21.3	268	24.9	13.7	7.1	28.6	197
19.1	16.2	1.4	8.2	318	22.6	11.9	7.3	29.3	170
					20.1	9.4	6.6	28.3	134
					17.7	9.7	6.3	22.0	134

Mixing chamber 2 1/2 " long

8.9	6.2	5.0	23.5	101	32.5	6.3	1.7	23.4	86
11.0	5.4	4.2	30.0	89	30.6	5.9	2.1	27.6	77
12.5	4.9	3.1	29.4	81	28.5	6.7	2.9	29.7	88
14.6	5.3	2.7	30.0	90	25.5	6.3	3.3	30.0	82
15.9	10.1	4.3	28.8	174	23.5	7.1	4.1	29.2	91
17.5	17.9	6.1	27.2	316	21.3	5.0	3.0	26.3	64

Mixing chamber 3" long

18.4	6.8	1.2	15.4	119	32.4	6.3	1.7	23.3	76
16.8	5.8	2.2	28.2	102	30.6	5.9	2.1	27.6	80
14.9	5.2	2.8	32.8	89	28.7	5.7	2.3	28.0	75
13.3	4.9	3.1	32.2	79	26.5	6.3	3.3	31.9	82
11.7	5.5	4.1	31.5	90	24.4	7.3	3.9	28.6	92
10.8	4.7	3.3	26.5	75	21.6	7.4	3.8	23.0	92

Mixing chamber 4" long

10.6	4.8	3.2	22.7	77	32.3	6.2	1.8	24.8	84
12.4	5.7	3.9	30.9	93	30.0	5.7	2.3	30.2	76
14.1	5.3	2.7	28.3	85	27.6	5.4	2.6	31.4	72
15.8	5.7	2.3	27.2	96	25.4	5.3	2.7	29.0	67
17.4	6.2	1.8	23.0	106	23.2	6.6	3.0	22.5	80
18.4	5.4	1.0	16.2	103	20.5	6.4	3.2	20.7	80

Mixing chamber 5" long

$p_e = 39.4$ #/□"					$p_e = 70.0$ #/□"				
p_f	$t\,Q_e$	$t\,Q_s$	η	t	p_f	$t\,Q_e$	$t\,Q_s$	η	t
20.2	6.4	0.0	0.0	105	21.6	6.3	3.3	23.4	81
18.3	4.0	0.8	17.3	74	24.2	7.4	3.8	27.1	94
16.6	5.9	2.1	25.9	102	26.4	5.5	2.5	27.6	71
14.7	8.5	4.3	30.1	146	28.6	6.9	2.7	29.1	92
12.9	4.9	3.1	30.8	84	30.5	6.0	2.0	25.8	82
10.7	5.8	3.8	24.4	93	31.7	7.7	1.9	20.4	105

Mixing chamber 6" long.

9.6	6.0	3.5	18.8	107	21.5	5.4	2.6	21.3	72
11.7	5.1	2.9	24.0	93	24.4	5.5	2.5	23.6	76
13.7	5.7	2.3	21.5	99	28.2	5.8	2.2	25.6	82
15.7	5.8	2.2	25.1	107	31.1	6.4	1.6	20.0	90
17.7	7.8	1.7	17.8	142	34.0	7.5	0.5	6.3	104
19.6	7.3	0.7	9.5	140					

TRAVERSES OF MINGLING STREAMS

In order to study the manner in which the inner and outer jets combine, I made a series of traverses with a Pitot tube at different distances from the nozzles. The first was very close, within about 1/32 of an inch of the tips. For the other traverses I attached mixing chambers of lengths varying from one to six inches and tested the speed at the open end. In every case the water spurted into the open air and was deflected downward into the weir box. Pressures behind the nozzles were kept constant at 50#/□" and 9.0 #/□" for all of these tests. The Pitot tube was moved across in steps averaging .02" each (less where the speed was varying), and the pressure within it read on a gauge at each step.

Taking the constant of the tube as .99, a value determined by Professor LeConte, the velocity at any point is $.99\sqrt{2g\,h}$, where h is the head in feet of water corresponding to p_p, gauge reading. Hence $v = .99\sqrt{64.4 \times p_p \times 144/62.4} = 12.07\sqrt{p_p}$

The velocities are plotted on curve sheets Nos. 1 to 3 inc.

Results of mingling stream traverses

It will be seen by reference to the curves that the streams act upon each other to a considerable extent within one inch of the nozzles, the inner part of the annular jet gaining speed lost by the outer part of the other. Friction against the mixing tube is seen to decrease the peripheral speed of the annular stream.

With each increase in length these effects become more marked, except that the higher speed is gradually communicated clear to the periphery of the outer jet and counteracts the friction on the walls. The rubbing velocity, which commenced at 25 ft. per second, drops to about 21 within the first 1 3/8 " and then rises gradually until it reaches 45 ft. per second at the end of six inches. Probably the deceleration of friction at the beginning would be considerably greater but for the fact that the outer jet contracts very much as it leaves the nozzle and so hardly touches the walls for a certain distance from that point. This can not be shown by the Pitot tube because of its imperfect action near the periphery. See also page 10.

Meanwhile the effect of the drag of the outer water is propagated toward the center of the driving stream. The rapid portion of this grows more and more slender to two inches from the nozzles. Here the center continues to keep the velocity with which it left its nozzle, 86 ft. per second. From this point onward the slower water decelerates it until the central speed becomes 65 ft. per second at six inches.

There is an irregularity to be noticed in all the later curves - that the speed is greater above the axis than at corresponding points below it. Possibly this is due to some small obstruction lodged in the lower part of the annular nozzle after one

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or two of the traverses had been made. I did not notice it at the time and after the curves were drawn did not have an opportunity of making any of the runs again.

LOSSES IN PARTS OF JET PUMP

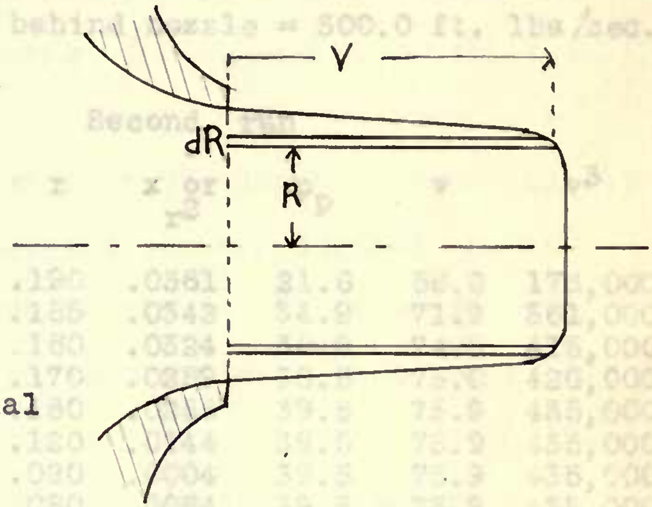
A considerable part of the work consisted in testing the inner and outer nozzles, mixing chambers, and diffuser to find their loss coefficients. In general this coefficient, ζ, is defined by: $\zeta\, v^2/2g$ = lost head(in feet of water) in the part considered. It is assumed that ζ is a constant for any piece of apparatus under varying conditions of velocity. I found in all cases that experimental results for each part, when plotted against v gave no significant trend either way, which shows that the exponent 2 in the expression $\zeta\, v^2/2g$ is correct within the limits of my work.

I supposed that there was no contraction of the inner jet and made a series of the ordinary tests to determine its coefficient, using various pressures behind the nozzle. The water issuing at each pressure was weighed for a measured time . The diameter of the nozzle is 3/8", giving an area = .000768 sq. ft. The velocity, v_j , is computed as Q/a.

h_e	t	Lbs.	CuFt Wt.	CuFt Meter	Q_e	v_j	$\frac{v_j^2}{2g}$	$1+\zeta$	ζ
185.7	180	1400	22.4	22.5	.0711	92.6	133.1	1.395	.395
161.5	242	1400	22.4	22.6	.0662	86.2	115.2	1.403	.403
90.7	317	1400	22.4	22.6	.0506	65.9	67.5	1.345	.345
35.0	459	1500	24.0	24.1	.0314	40.9	26.0	1.345	.345

The impossible magnitude of the ζ thus obtained proves that the jet must contract. I made two other tests using the Pitot tube to determine the velocity of the water in every part of the stream, measuring the Pitot pressure once with a mercury

column and the other time with a pressure gauge. As shown on page 6 the velocity $v = 12.07\sqrt{p_p}$. Similarly $v = .99\sqrt{2g \times 13.6\, h_m/12}$ $= 84.55\sqrt{h_m}$, where h_m indicates the inches of mercury - of course corrected for water column.



In the appended figure let v be the component of the velocity of the stream parallel to the axis of the nozzle at a point distant R feet from it. Let dR represent the thickness of a differential hollow cylinder of water of radius R and length v. The weight of the water $= 2\pi R\, dR\, v\, \gamma$, where γ is the number of pounds per cubic foot. The integral of this from R = 0 to R = the outer radius of the jet would be the weight of water leaving the nozzle in one second.

The kinetic energy of the differential cylinder = weight $\cdot\, v^2/2g = \frac{\pi\gamma}{g} R\, dR\, v^3$. The total energy of the water leaving the nozzle in one second is the integral of this from center to circumference. If we measure the variable radius in inches (=r), the expression becomes $\int \frac{\pi\gamma}{g} \frac{r}{12} \frac{dr}{12} v^3$. Now let $x = r^2$; then $dx = r \cdot dr \cdot 2$. Substituting: Ft. Lbs. per second of jet $= \frac{\pi\gamma}{288\, g} \int_0^{r_j} v^3\, dx$

The integral can be evaluated by plotting v^3 against x $(= r^2)$. The curves will be found on curve sheet No.4 and a tabulation of most of the data and computation is on page 10.

For the first of the runs the mercury column was used. Pressure $p_e = 23.0$ #/□" ; $Q_e = .0399$; $Q_e/a_j = 52.0$ ft. per sec. For velocity head, $a_e = .0104$ □ ft. (1 1/4 " pipe at pressure gauge). Hence $v_e = 3.84$ and velocity head = .23 ft. Pressure head = 53.1 ft.

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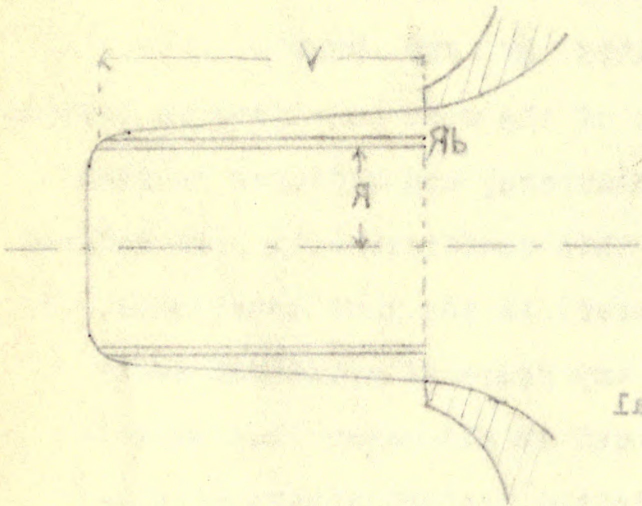

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The kinetic energy of the differential cylinder = weight x $v^2/2g = \frac{\pi\gamma}{g} R\, dR\, v^3$. The total energy of the water leaving the nozzle in one second is the integral of this from center to circumference. If we measure the variable radius in inches (=r), the expression becomes $\int \frac{\pi\gamma}{g} \frac{r}{12} \frac{dr}{12} v^3$. Now let $x = r^2$; then $dx = r \cdot dr \cdot 2$. Substituting: Ft. lbs. per second of jet $= \frac{\pi\gamma}{288\, g} \int_0 v^3\, dx$

The integral can be evaluated by plotting v^3 against $x\ (= r^2)$. The curves will be found on curve sheet No.4 and a tabulation of most of the data and computation is on page 10.

For the first of the runs the mercury column was used. Pressure $p_e = 23.5$ #/□" ; $Q_e = .0399$; $Q_e/a_j = 58.0$ ft. per sec. For velocity head, $a_e = .0104$ □ ft. (1 1/4 " pipe at pressure gauge). Hence $v_e = 3.84$ and velocity head = .23 ft. Pressure head = 53.1 ft.

The energy behind the nozzle, then, = .0399 · 62.4 · 53.3 = 132.6 foot lbs. per second.

For the second run: h_e = 91.4; Q_e = .0523; v_e = 5.03 and velocity head = .39; energy behind nozzle = 300.0 ft. lbs/sec.

First run

r	x or r^2	h_m	v	v^3
.190	.0361	15.7	33.5	37,500
.185	.0342	33.5	48.9	117,000
.180	.0324	43.9	56.0	176,000
.170	.0289	47.0	57.9	194,000
.150	.0225	47.2	58.0	196,000
.120	.0144	47.4	58.2	197,000
.070	.0049	47.6	58.3	198,000
.020	.0004	47.2	58.0	196,000
.030	.0009	47.1	58.0	195,000
.060	.0036	46.8	57.8	192,000
.080	.0064	46.8	57.8	192,000
.130	.0169	46.9	57.9	194,000
.160	.0256	46.5	57.6	191,000
.180	.0324	46.1	57.4	190,000
.185	.0342	45.1	56.7	182,000
.190	.0361	24.9	42.1	75,000

Second run

r	x or r^2	p_p	v	v^3
.190	.0361	21.6	56.0	176,000
.185	.0342	34.9	71.2	361,000
.180	.0324	38.2	74.5	415,000
.170	.0289	38.5	75.0	420,000
.160	.0256	39.5	75.9	435,000
.120	.0144	39.5	75.9	435,000
.020	.0004	39.5	75.9	435,000
.080	.0064	39.5	75.9	435,000
.180	.0324	39.1	75.5	430,000
.185	.0342	35.4	71.7	370,000
.190	.0361	24.4	59.5	211,000

Second run: area = 151.4 □ cm = 15,140 units; average ordinate = mean v^3 = 430,000

First run: area = 67.28 □ cm = 6,728 units; average ordinate = mean v^3 = 191,100.

Inserting the area of the curves in the expressions for kinetic energy on page 9 gives 142.3 and 320.0 ft. lbs per sec respectively, values in excess of the total energy behind the nozzle. This is, of course, absurd. The trouble is due to the fact that the jet really contracts but the Pitot tube erroneously indicates a considerable velocity up to and beyond the point where its center is opposite the edge of the nozzle (r = 3/16 " = .1875) as shown in the tables above. This is because the water received when the tube is partly in the air makes a pressure within the apparatus.

To avoid the difficulty I have taken v_j = the

Pitot tube traverses of inner nozzle

The energy behind the nozzle, then, = .0399 · 62.4 · 53.3 = 132.6 foot lbs. per second.

For the second run: h_e = 91.4; Q_e = .0523; v_e = 5.03 and velocity head = .39; energy behind nozzle = 300.0 ft. lbs/sec.

First run					Second run				
r	x or r^2	h_m	v	v^3	r	x or r^2	p_p	v	v^3
.190	.0361	15.7	33.5	37,500	.190	.0361	21.6	56.0	176,000
.185	.0342	35.6	48.9	117,000	.185	.0342	34.9	71.2	361,000
.180	.0324	45.9	56.0	176,000	.180	.0324	38.3	74.5	415,000
.170	.0289	47.0	57.9	194,000	.170	.0289	38.5	75.0	420,000
.150	.0225	47.2	58.0	195,000	.160	.0256	39.5	75.9	435,000
.120	.0144	47.4	58.2	197,000	.120	.0144	39.5	75.9	435,000
.070	.0049	47.6	58.3	198,000	.020	.0004	39.5	75.9	435,000
.020	.0004	47.3	58.0	195,000	.080	.0064	39.5	75.9	435,000
.030	.0009	47.1	58.0	195,000	.180	.0324	39.1	75.6	430,000
.060	.0036	46.8	57.8	193,000	.185	.0342	35.4	71.7	370,000
.080	.0064	46.8	57.8	193,000	.190	.0361	24.4	59.5	211,000
.130	.0169	46.9	57.9	194,000					
.160	.0256	46.5	57.6	191,000					
.180	.0324	46.1	57.4	190,000					
.185	.0342	45.1	56.7	182,000					
.190	.0361	24.9	42.1	75,000					

Second run: area = 151.4 □ cm = 15,140 units; average ordinate = mean v^3 = 450,000

First run: area = 67.28 □ cm = 6,728 units; average ordinate = mean v^3 = 191,100.

Inserting the area of the curves in the expression for kinetic energy on page 9 gives 142.3 and 320.0 ft. lbs per sec respectively, values in excess of the total energy behind the nozzle. This is, of course, absurd. The trouble is due to the fact that the jet really contracts but the Pitot tube erroneously indicates a considerable velocity up to and beyond the point where its center is opposite the edge of the nozzle (r = 3/16" = .1875) as shown in the tables above. This is because the water received when the tube is partly in the air makes a pressure within the apparatus.

To avoid the difficulty I have taken v_j = the

cube root of the average ordinate of the curve, considering it the "effective velocity" in analogy to the use of the word effective in electrical calculations. This value is subject to some error in that the curve area, taken for the full radius of the nozzle, is wider than it should be, and the diminishing velocity near the periphery is too large. The result is that the area of the curve is considerably too large, accounting for the absurd result on page 10, but the abscissal distance divided into this area to obtain the mean ordinate is also in excess, tending to bring the mean ordinate back toward the correct value.

As a check I plotted the velocity against r^2 for these two runs and found the average velocity in each case exactly identical with the effective value. This shows that with so nearly uniform speed across the jet as is usually given with nozzles it is unnecessary to take the trouble to find the effective velocity.

The values of v_j as found were 57.6 and 75.5 feet per second respectively for the two runs, giving 51.5 and 88.5 feet as the velocity heads. Dividing into 53.3 and 91.8 (the total - pressure and velocity - heads behind the nozzle) , gives 1.035 and 1.037 as $1 + \zeta$ in each case.

Another run, not recorded above and not worked through with the v^3/r^2 plot, gave $\zeta = .045$. Giving greater weight to the first two, I take the average value of $\zeta_j = .037$

Let a_{oj} = the contracted area of the jet, defined by Q_j/v_j. The ratio $a_{oj}/a_j = \alpha_j$, the coefficient of contraction, where a_j is the nozzle area. With this nozzle the three runs give .000693, .000693, and .000692 = a_{oj} . Hence $\alpha_j = 693/768 = .902$.

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Let a_{oj} = the contracted area of the jet, defined by Q_j/v_j. The ratio $a_{oj}/a_j = \alpha_j$, the coefficient of contraction, where a_j is the nozzle area. With this nozzle the three runs give .000695, .000693, and .000693 = a_{oj} . Hence α_j = 693/763 = .908.

Friction and contraction coefficients, outer nozzle

To determine the coefficients of the annular nozzle I made one traverse with the Pitot tube, measuring quantity and pressures as before, except that instead of weighing the water I sent it into a measuring tank. The v^3/r^2 curve will be found on curve sheet No. 5 . The positions of the boundaries of the outer nozzle are shown there, and it may be seen that the stream is very much contracted and converges into the space supposed to be occupied by the central jet. It was difficult to determine the width to use in getting the area and mean ordinate, but by several applications of the cut and try method I secured a fair approximation to a satisfactory value. The trials were made by estimating the width, measuring the curve area and computing the average v, multiplying by the annular area corresponding to the width selected, and comparing with the measured Q_s. I finally took $v_r = 43.2$ as the best average, using both the v^3/r^2 curve and a v/r^2 curve (the latter not preserved). This gives the velocity head = 29.0. The pressure head back of the nozzle = 29.8 feet (12.9 #/□" on gauge), and the velocity head was neglected. Hence $1 + \zeta_r = 1.027$. Since an inaccuracy of even .1# in the gauge would change the value of ζ_r thus derived to .018 or .035 , the result is not very reliable. It is unimportant, however, for as will be shown later (page 18) the loss in this nozzle is almost zero compared with the other losses in the jet pump.

Q_s in this test was .04461. Taking $v_r = 43.2$ we find $a_{or} = .04461/43.2 = .001033$ □ ft. The nozzle area, a_r, = .00136; hence $\alpha_r = 1033/1360 = .76$. This is much more accurate than the ζ value, for it does not depend at all upon the pressure measurement. Of course, a_{or} indicates the area of the contracted stream leaving the annular nozzle $= Q_s / v_r$.

Friction and contraction coefficients, outer nozzle

To determine the coefficients of the annular nozzle I made one traverse with the Pitot tube, measuring quantity and pressures as before, except that instead of weighing the water I sent it into a measuring tank. The v^b/r^2 curve will be found on curve sheet No. 5 . The positions of the boundaries of the outer nozzle are shown there, and it may be seen that the stream is very much contracted and converges into the space supposed to be occupied by the central jet. It was difficult to determine the width to use in getting the area and mean ordinate, but by several applications of the cut and try method I secured a fair approximation to a satisfactory value. The trials were made by estimating the width, measuring the curve area and computing the average v, multiplying by the annular area corresponding to the width selected, and comparing with the measured Q_s. I finally took v_r = 43.2 as the best average, using both the v^b/r^2 curve and a v/r^2 curve (the latter not preserved). This gives the velocity head = 29.0. The pressure head back of the nozzle = 29.8 feet (12.9 #/□" on gauge), and the velocity head was neglected. Hence $1 + \zeta_r$ = 1.027. Since an inaccuracy of even .1# in the gauge would change the value of ζ_r thus derived to .018 or .035 , the result is not very reliable. It is unimportant, however, for as will be shown later (page 18) the loss in this nozzle is almost zero compared with the other losses in the jet pump.

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Method of getting mixing chamber coefficient

Since a number of mixing chambers of different lengths are used with this jet pump, a more convenient form for the friction coefficient is λ instead of ζ. The symbol is defined by: lost head (in feet of water) = $\lambda l/d \cdot v^2/2g$, where l/d is the ratio of the length considered to the diameter - of course applying only to cylindrical bore. For any definite length $\zeta = \lambda l/d$. For a uniform flow of water between the points 1 and 2 of a cylindrical tube, the speed of the water being constant throughout the area except as impeded at the periphery by skin friction, $h_1 + v^2/2g = h_2 + v^2/2g + \lambda l/d \cdot v^2/2g$; hence $\lambda l/d = (h_1 - h_2) \div v^2/2g$.

The method of testing was to connect several of the mixing chambers in series between the nozzles and diffuser and send water through from both orifices at once under equal pressures. To escape the complications at the contracted jet and to allow the water to achieve a uniform speed across the cross section, the measurements were always made at least six or eight inches from the nozzle plane. Water and mercury manometers were used to measure the drop in head across all possible different combinations of mixing chambers, and the water was weighed and the time measured for each run. One side of the manometer was always connected to the throat of the diffuser and the other side connected to the ends of the different chambers, a small brass tube being inserted in orifices in the flanges which communicated with the interior. Because the aperture in the throat was located about 3/8 " from the end of the last chamber it was necessary to add this length (which is that of the cylindrical portion of the diffuser) to the length of mixing tube measured. $a_m = .00213\square$ '.

Following is a table of a few of the tests:

Mixing chamber and diffuser coefficients

Water Lbs.	t secs.	Q	v	$\frac{v^2}{2g}$	Length inches	(h_1-h_2) Ft. Water	$\lambda l/d$	λ
1100	700	.0251	11.8	2.17	8.75	.708	.326	.0233
					17.25	1.383	.637	.0231
1500	483	.0497	23.3	8.46	2.75	1.033	.122	.0277
					4.75	1.475	.174	.0230
1500	366	.0656	30.8	14.72	2.75	1.575	.107	.0243
					4.75	2.520	.171	.0225
					8.75	4.100	.278	.0199
					17.25	7.880	.535	.0193

Omitting tests with very short lengths (under two inches), which gave inconsistent results, the values found for λ are shown in the following table. The table on the right gives values of ζ for all the different mixing chambers, computed from the mean $\lambda = .0243$ by ζ per inch length $= \lambda \div 5/8 = .0389$.

Length inches	velocities used	Mean λ	Mixing Chamber	Cylindrical length	ζ_m
2.75	23 - 31	.0260	0	.5 in.	.020
4.25	13 - 20 - 26	.0234	1 in.	1.5	.058
4.75	12 - 23 - 31	.0236	1 3/8	1.875	.073
6.25	13 - 20	.0262	2	2.5	.097
8.75	12 - 31 - 35	.0240	2 1/2	3.0	.117
17.25	12 - 31 - 35	.0226	3	3.5	.136
			4	4.5	.175
	Final Average	.0243	5	5.5	.214
			6	6.5	.253

In order to find whether λ depended upon the speed of the water I plotted all the values against v, and found the curve, though exceedingly irregular, roughly parallel to the axis of v.

To find the coefficient of the diffuser, water was run through it at measured rates and the difference in pressure between the two ends read on a mercury column. The water came through both nozzles at the same pressure and several mixing chambers were inserted to give it time to get over the effects of contraction and impact. The loss in head is refferred to the higher velocity, v_t, and $= \zeta_d v_t^2/2g = v_t^2/2g - v_f^2/2g - (h_f - h_t)$. Following is a table

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showing the values obtained in eight different runs:

v_t	13.0	20.3	21.8	26.0	29.7	30.7	30.7	33.7
ζ_d	.138	.120	.135	.113	.111	.110	.110	.116

The average of all these values is $\zeta_d = .119$ or .12

Two other runs under similar conditions gave answers so widely different as to show error in measurement or irregular action and their results were thrown out.

ANALYSIS OF FRICTION IN MIXING CHAMBER

It is customary to express the head lost due to friction in the mixing chamber as $v_t^2/2g \cdot \zeta_m$, where v_t is the average value of the speed of the water at the throat of the diffuser, or Q_f/a_t . This is considerably in excess of the true loss of head, for the rubbing velocity is variable through the tube, never becoming so great as v_t and approaching it only near the end. It is theoretically v_r at the plane of the nozzles, assuming no contraction. Hence a nearer approach to exactness would be to substitute for v_t in the expression above an average velocity = $(v_r + v_t)/2$. This gives head lost = $\zeta_m (v_r + v_t)^2/8g$. It should be noted in this connection that any measurement of ζ such as recorded on pages 13 and 14 must be made with constant rubbing velocity. The term 'rubbing velocity' as used here refers not to the actual speed of the water particles in contact with the wall of the tube, but to the velocity of the major part of the water near it - a value corresponding to the velocity obtained in the tests mentioned by dividing Q by a_t.

Following is the derivation of an expression for the loss of head here, which is perhaps more accurate, at least theoretically, than those mentioned above. No account is taken of the

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contraction of the jets and it is assumed that the rubbing velocity varies uniformly from end to end of the mixing chamber, which seems to be in substantial accord with the facts.

Referring to Fig. 2 on Sketch Plate No. 1, in the differential distance dl there is a loss of head = $\lambda \cdot dl/d \cdot v^2/2g$ where v represents the rubbing velocity and $= v_r + (v_t - v_r) \cdot l/l_m$. The total head lost in the tube is the integral of the differential loss between $l = 0$ and $l = l_m$, or $\lambda/2dg \cdot \int_0^{l_m} v_r^2\, dl + 2v_r(v_t - v_r)/l_m \cdot l\, dl + (v_t - v_r)^2/l_m^2 \cdot l^2\, dl = \lambda/2dg \left[v_r^2 l_m + 2v_r(v_t - v_r)/l_m \cdot l_m^2/2 + (v_t - v_r)^2/l_m^2 \cdot l_m^3/3 \right] = \zeta_m/6g \cdot (v_r^2 + v_r v_t + v_t^2)$ since $\lambda l_m/d = \zeta_m$. This is the head lost in the mixing chamber, that is, the amount by which the water reaches the throat robbed of the pressure it would have had if there were no friction. Hence the energy at that point is less than the no-friction value by the quantity: $Q_f \gamma \cdot \zeta_m/6g \cdot (v_r^2 + v_r v_t + v_t^2)$, which is the loss of energy per second through the tube.

To compare the result thus obtained with the other formulas mentioned previously, I have computed the loss in each of the three ways in each of five runs made with the six inch mixing chamber. The data and computation follow:

Run	Q_f	v_r	v_t	$Q_f\ v_t^2/2g$	$Q_f\ (v_r+v_t)^2/8g$	Last Way
1	.111	35.0	52.1	73.8	51.5	52.3
2	.1052	31.2	49.4	63.6	41.9	42.6
3	.0976	25.9	45.8	50.2	30.7	31.6
4	.0889	17.2	42.2	39.2	19.2	20.4
5	.0769	4.6	36.1	24.5	7.8	9.3

It is seen that the losses "last way" and using the average velocity are in close agreement and much lower than the value obtained in the usual way.

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$l\, dl + (v_t - v_r)^2/l_m^2 \cdot l^2\, dl \;=\; \lambda/2dg \left[v_r^2\, l_m + 2v_r(v_t - v_r)/l_m \cdot l_m^2/2 + (v_t - v_r)^2/l_m^2 \cdot l_m^3/3 \right] = \zeta_m/6g \cdot (v_r^2 + v_r v_t + v_t^2)$ since $\lambda\, l_m/d = \zeta_m$. This is the head lost in the mixing chamber, that is, the amount by which the water reaches the throat robbed of the pressure it would have had if there were no friction. Hence the energy at that point is less than the no-friction value by the quantity: $Q_r \gamma \cdot \zeta_m/6g \cdot (v_r^2 + v_r v_t + v_t^2)$, which is the loss of energy per second through the tube.

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Professor Hesse's expression for impact loss

IMPACT LOSSES

All the friction losses in the jet pump are not enough to account for its low efficiency. The additional loss is considered to be due to impact and eddying in the mixing chamber, and in the analysis as developed by Professor Hesse it has been expressed by the same formula as that used for the loss due to sudden enlargement in pipe section. Both the suction water and that entering through the central nozzle are considered to suffer loss of energy in this way, making the total diminution of energy = $Q_e \gamma (v_j - v_t)^2/2g + Q_s \gamma (v_t - v_r)^2/2g$, as for sudden enlargement.

In order to test the reliability of this expression and at the same time secure as thorough a check as possible on all the work described in this thesis, I have computed the entire losses for the series of five runs made with the six inch chamber with 70#/□" pressure on the inner nozzle. The data will be found on page 6. Below are the work and results. It will be seen that the sum of the computed losses of friction, impact and eddying is almost exactly equal to the difference between output and input. Velocity heads are allowed for for both the delivery water and that entering the central nozzle, and the work has been checked over to insure accuracy in the mathematics.

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See the following page for the tabulation:

Input, output, and losses computed. Energy balance.

Run	Time secs.	tQ_e	Q_e	Q_e/a_{oj} (v_j)	$Q_e\gamma\zeta_j$ $v_j^2/2g$	tQ_s	Q_s	Q_s/a_{or} (v_r)	$Q_s\gamma\zeta_r$ $v_r^2/2g$
1	72	5.4	.0750	108.2	31.4	2.6	.0361	35.0	1.3
2	76	5.5+	.0730	105.3	29.0	2.5-	.0322	31.2	.9
3	82	5.8	.0707	102.0	26.3	2.2	.0268	25.9	.5
4	90	6.4	.0711	102.6	26.8	1.6	.0178	17.2	.2
5	104	7.5	.0722	104.1	28.0	0.5	.0048	4.6	.003

Run	tQ_f	Q_f	Q_f/a_t (v_t)	$Q_f\gamma\zeta_d$ $v_t^2/2g$	$v_r^2+v_rv_t+v_t^2$	$Q_f\gamma\zeta_m(--)/6g$
1	8.0	.1110	52.1	35.0	5,763	52.3
2	8.0	.1052	49.4	29.3	4,954	42.4
3	8.0	.0976	45.8	23.8	3,956	31.6
4	8.0	.0889	42.2	18.4	2,802	20.4
5	8.0	.0769	36.1	11.7	1,483	9.3

Run	$(v_j-v_t)^2$ $Q_e\gamma/2g$	$(v_t-v_r)^2$ $Q_s\gamma/2g$	p_f	$h_f+v_f^2/2g$	$Q_s\gamma(--)$ Output	$h_e+v_e^2/2g$ $-(h_f+v_f^2/2g)$	$Q_e\gamma(-)$ Input
1	228	10.2	21.5	50.1	112.8	112.2	526
2	213	10.3	24.4	56.8	114.1	105.5	480
3	217	10.3	28.2	65.4	109.5	96.9	427
4	252	10.8	31.1	72.1	80.1	90.2	400
5	324	4.6	34.0	78.7	23.6	83.6	377

Run	Input - Output	Sum computed losses
1	413	358
2	366	325
3	317	309
4	320	329
5	353	378

CONCLUSION

The results of the investigation have been:

With constant applied pressure from thirty-nine to seventy pounds per square inch above atmosphere, the efficiency of this jet pump reaches a maximum when the delivery pressure is about thirty-four per-cent of that applied, the suction pressure being atmospheric . Comparing mixing chambers shows that under these conditions the greatest efficiency - about thirty-two per-cent, is secured by using the three inch chamber. However, any length from one and three-eights inches to four gives almost as good operation. All lengths outside of these show much lower efficiency.

The Pitot tube traverses show that, with fifty and nine pounds respectively behind the inner and outer nozzles the jets mingle in a distance of six inches so that the highest velocity in the center is less than three-halves that of the slowest water. Up to this point the outer jet is continually accelerated and the inner one retarded.

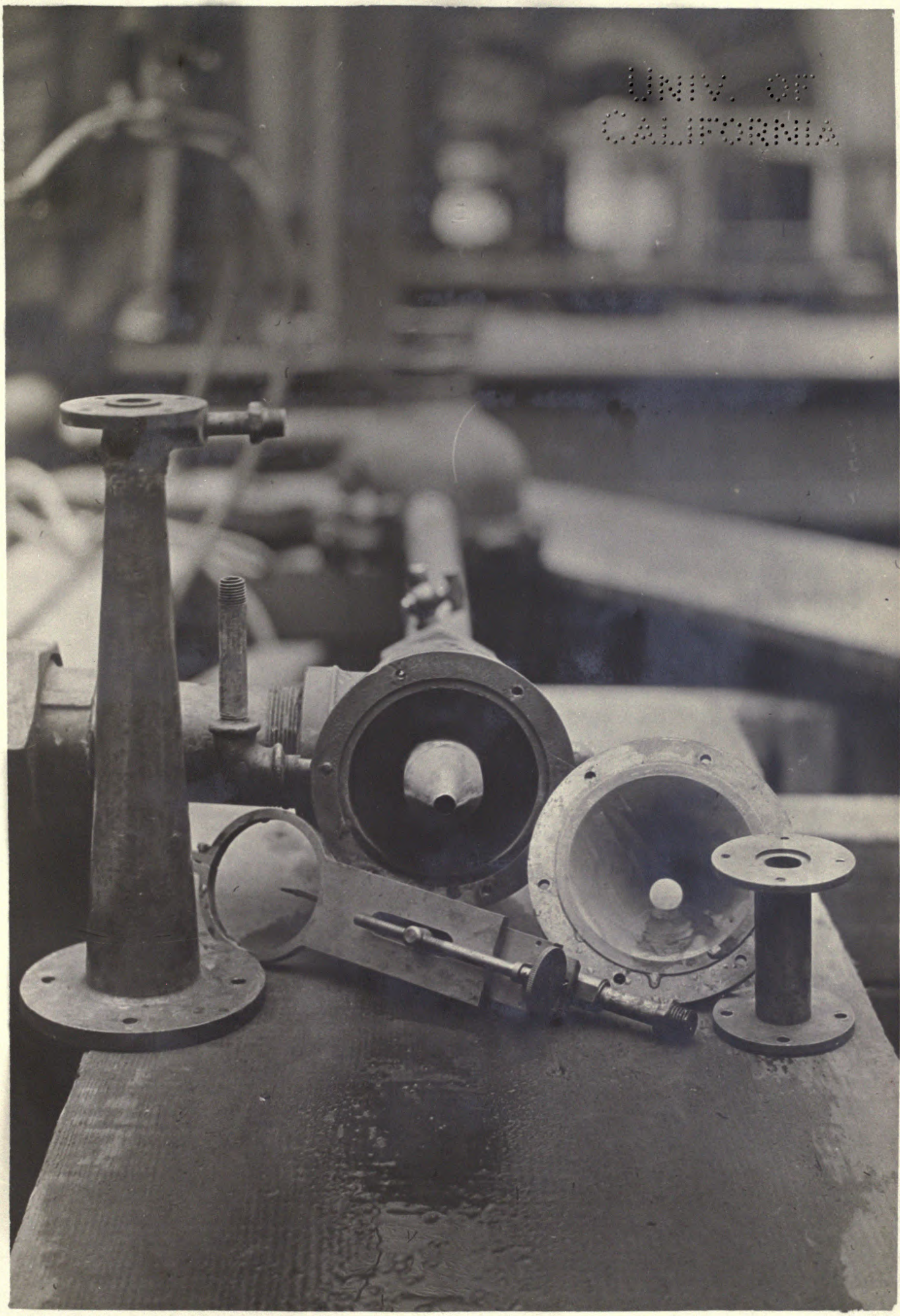
The loss coefficients of the parts of the pump are: Inner nozzle, .037; outer nozzle, .03; mixing chamber, .0389 per inch length; diffuser .12. The coefficient of contraction of the inner nozzle =.902 and of the outer nozzle .76

The expression derived by integration for the friction loss in the mixing chamber has been checked numerically on a reasonable approximation.

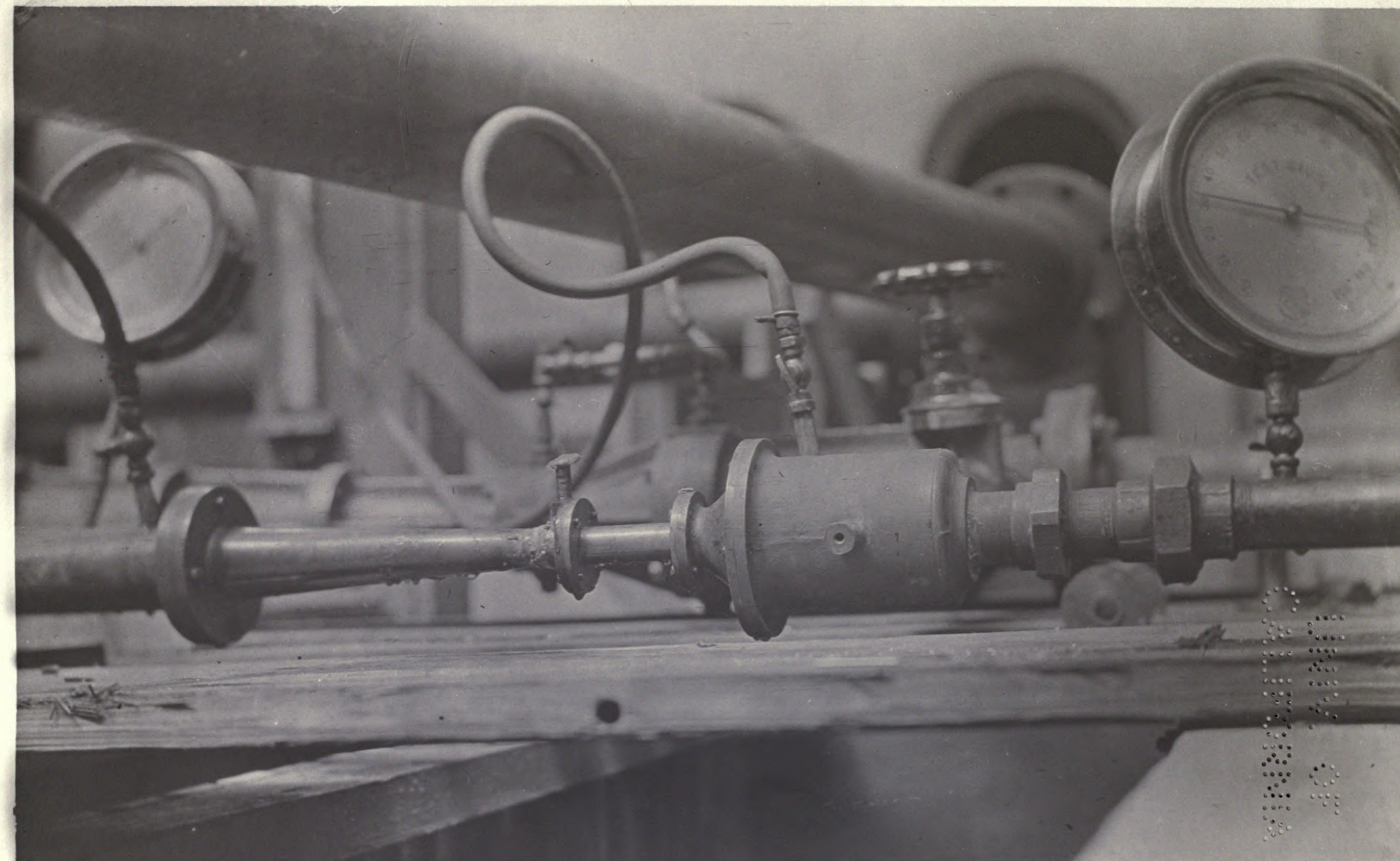
Professor Hesse's expression for the loss due to impact has been tested by substituting values from five efficiency runs and shown to agree very closely with the loss not accounted for by friction. Incidentally a rough check was secured on all the

coefficients found and the mixing chamber formula. However, since impact represents over two-thirds of the total loss, the latter check has little importance.

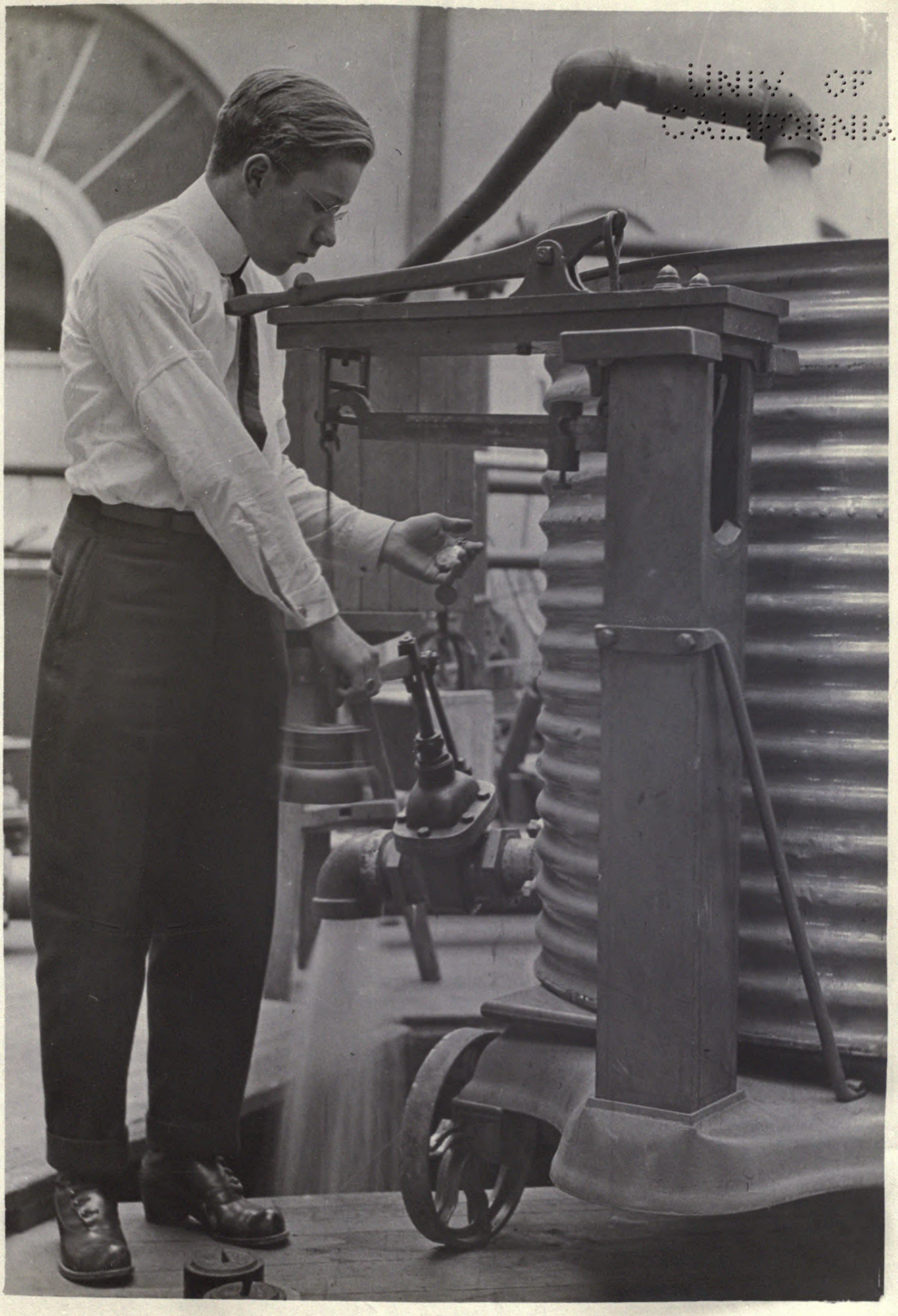
I would suggest that a further study of the impact loss with this apparatus would probably prove valuable. The data tabulated in this report would furnish material for determining the loss by impact under conditions of considerable variation, and perhaps point the way to the discovery of a more accurate expression for this very important factor in jet pump operation.



The Jet Pump Disassembled

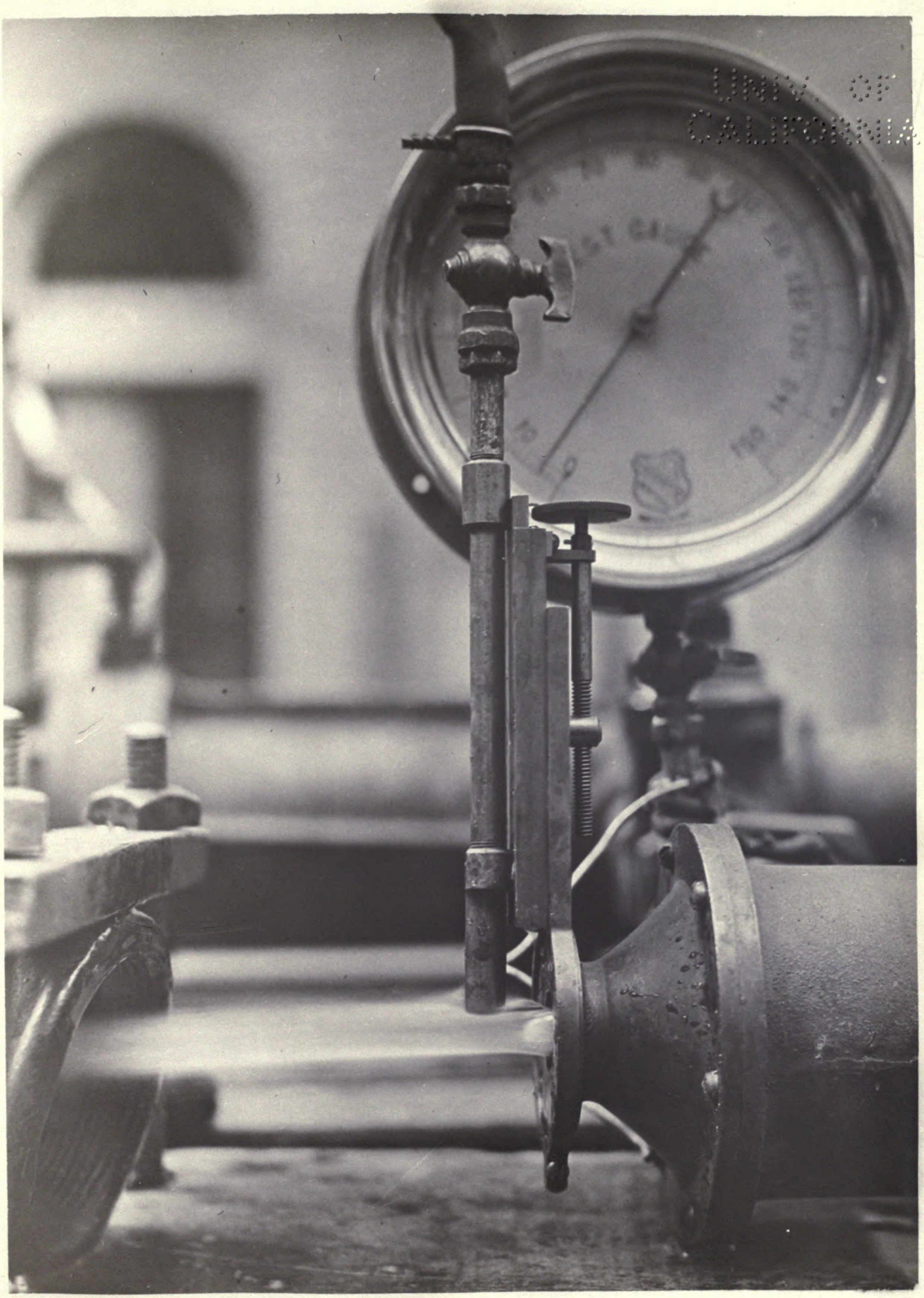


Connections for Efficiency Tests



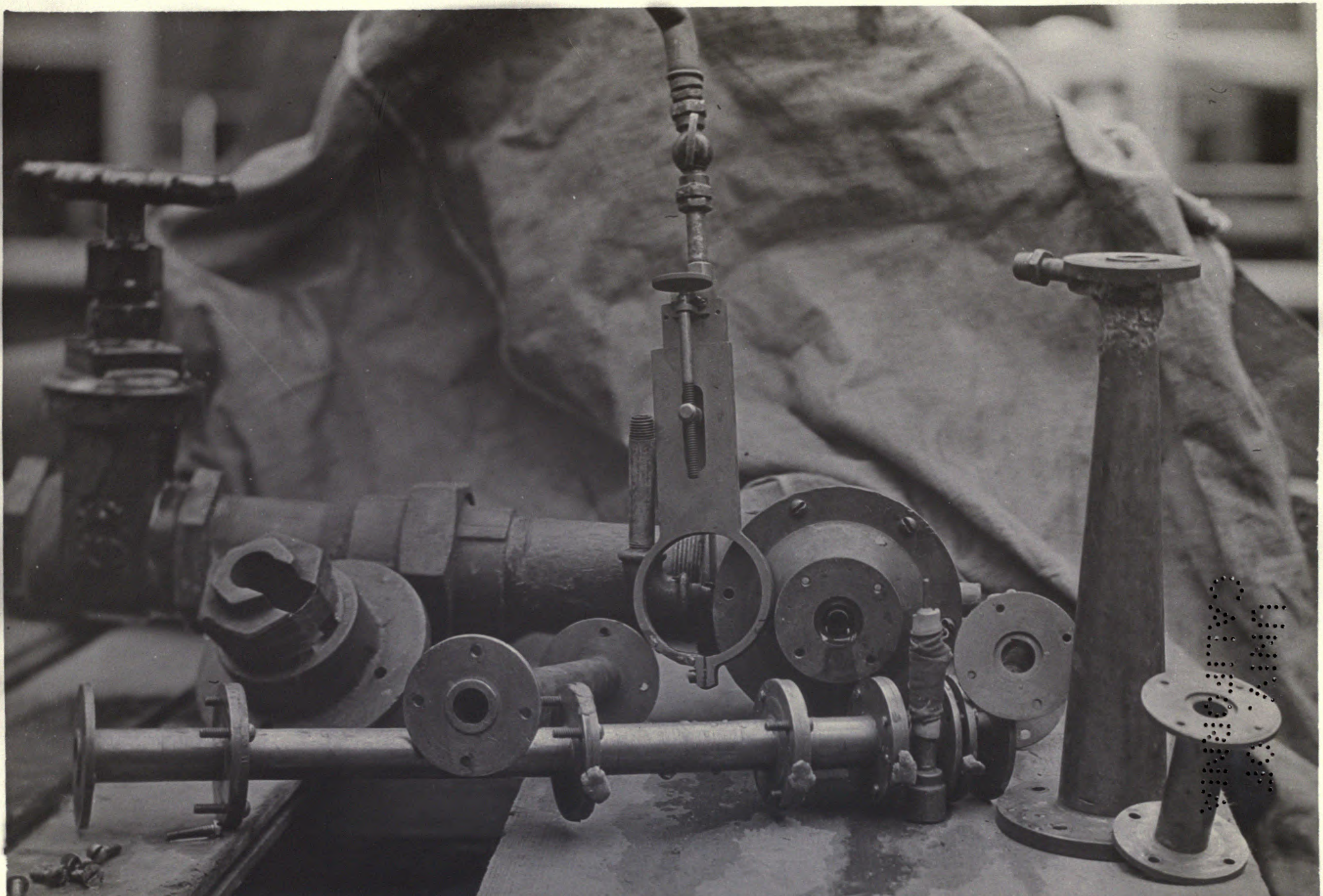
Assistant Measuring Weight and Time to Determine Q_f

309

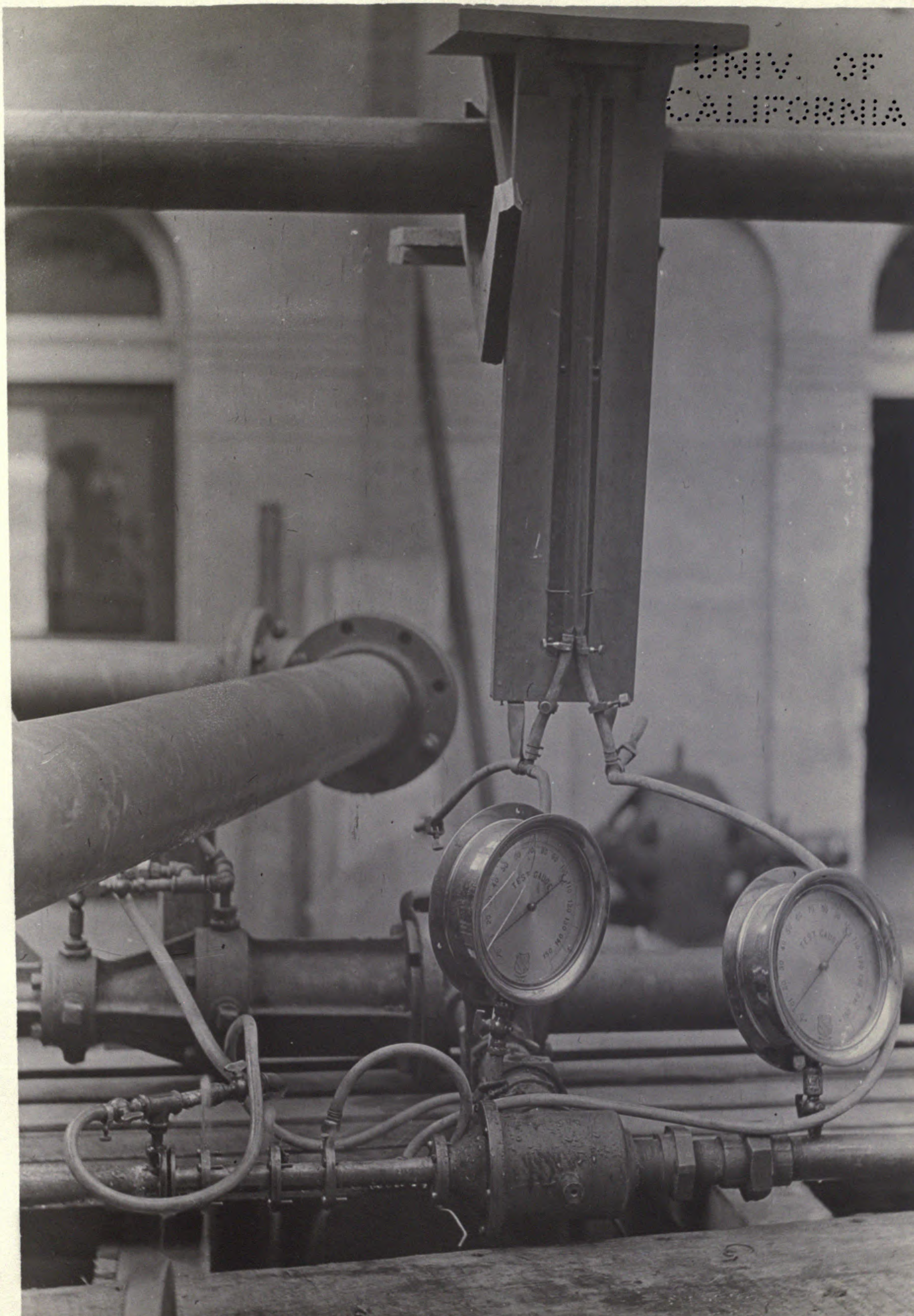


Traverse at Nozzle with Pitot Tube and Mercury Column

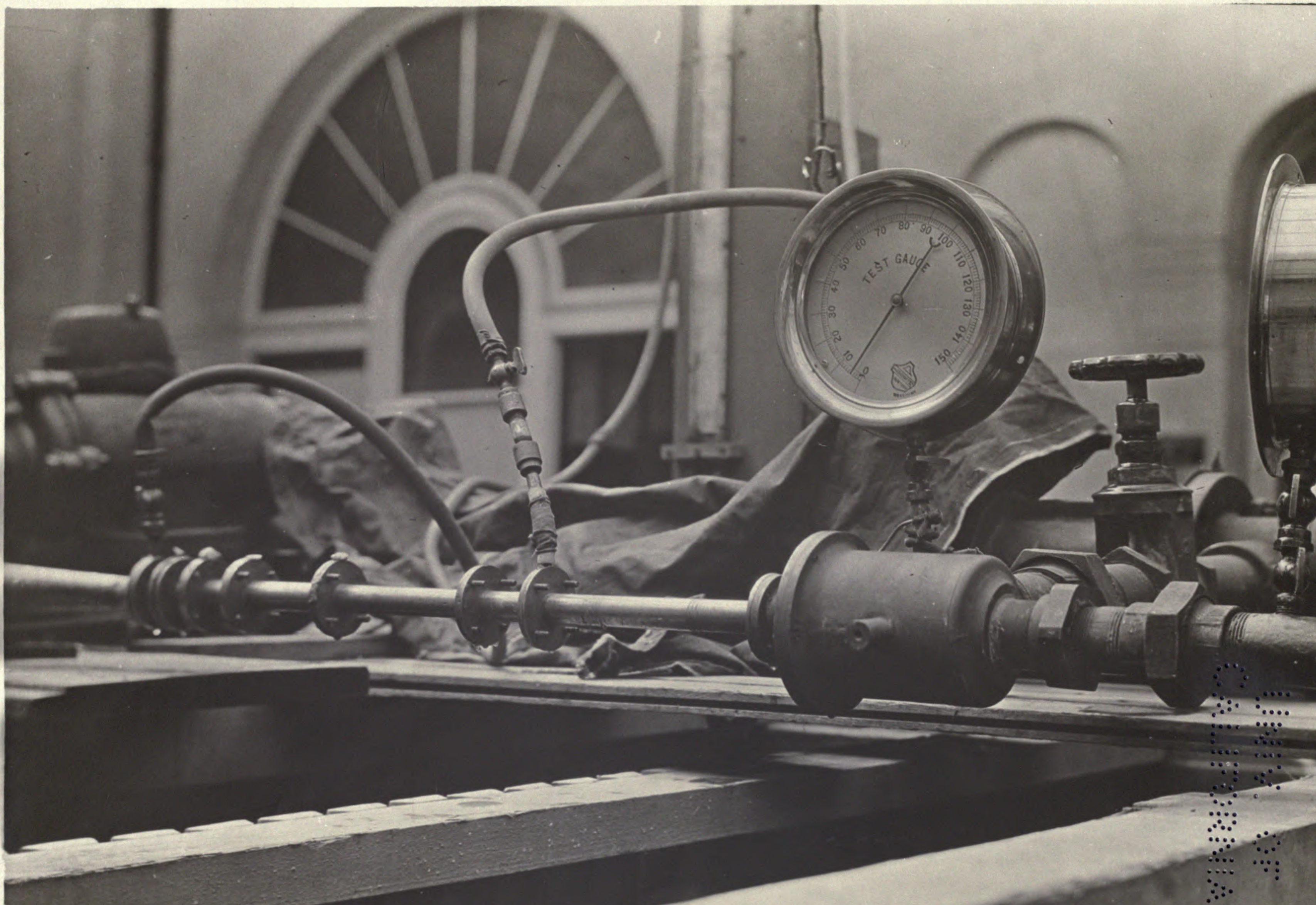
Pitot Tube Traverse at End of Mixing Chamber



All Parts of the Pump and Some other Apparatus



Connections to Inverted water Column for
Testing Mixing Chamber Coefficients



Connections to Mercury Column for Testing
Mixing Chamber Coefficients

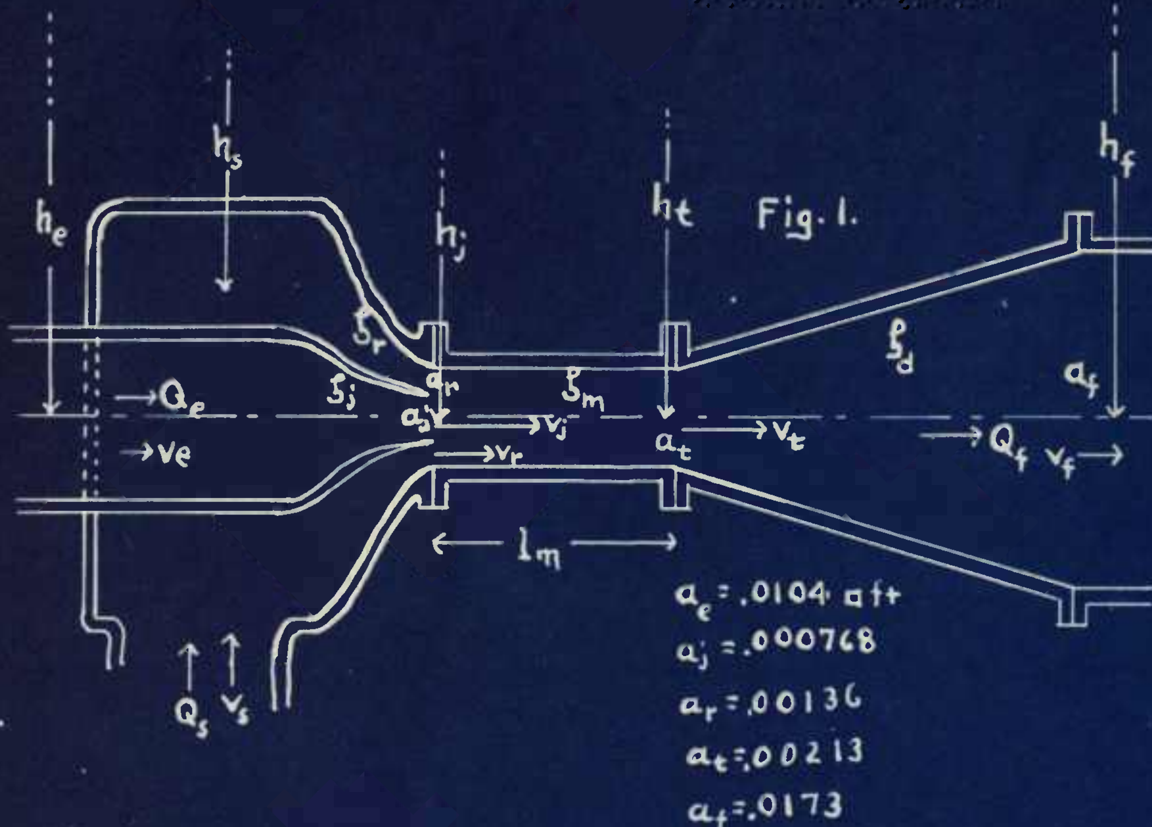


Fig. 1.

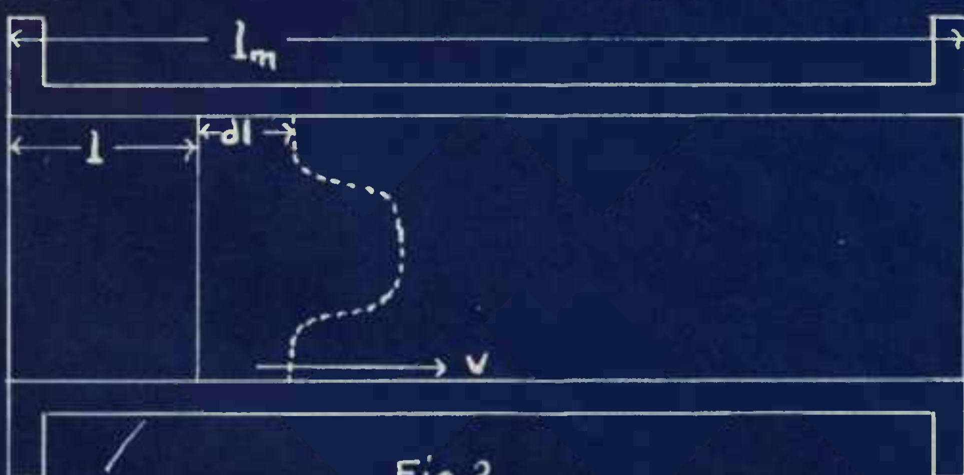
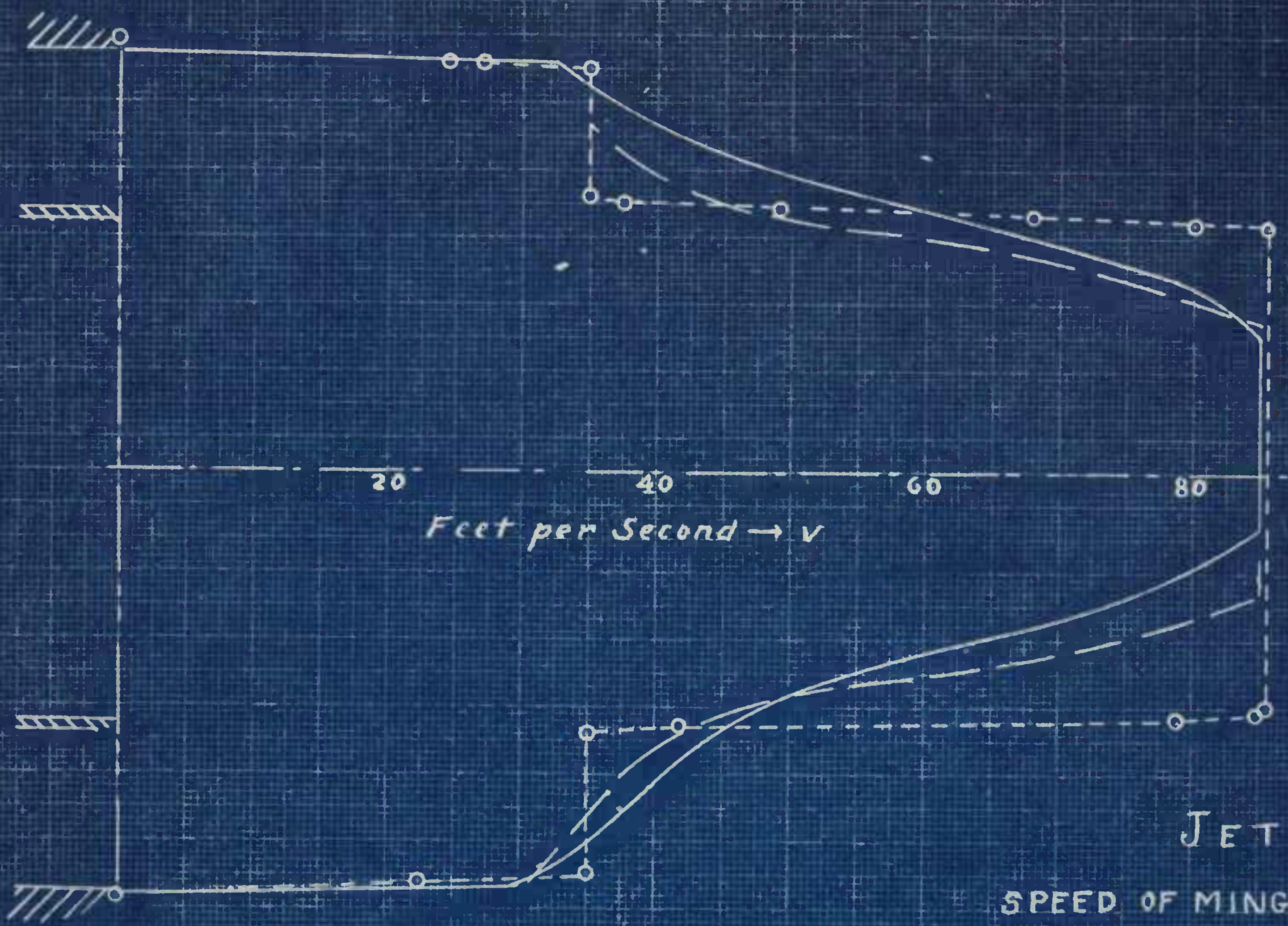


Fig. 2.

SKETCH PLATE No. 1.

H. H. Bliss



JET PUMP

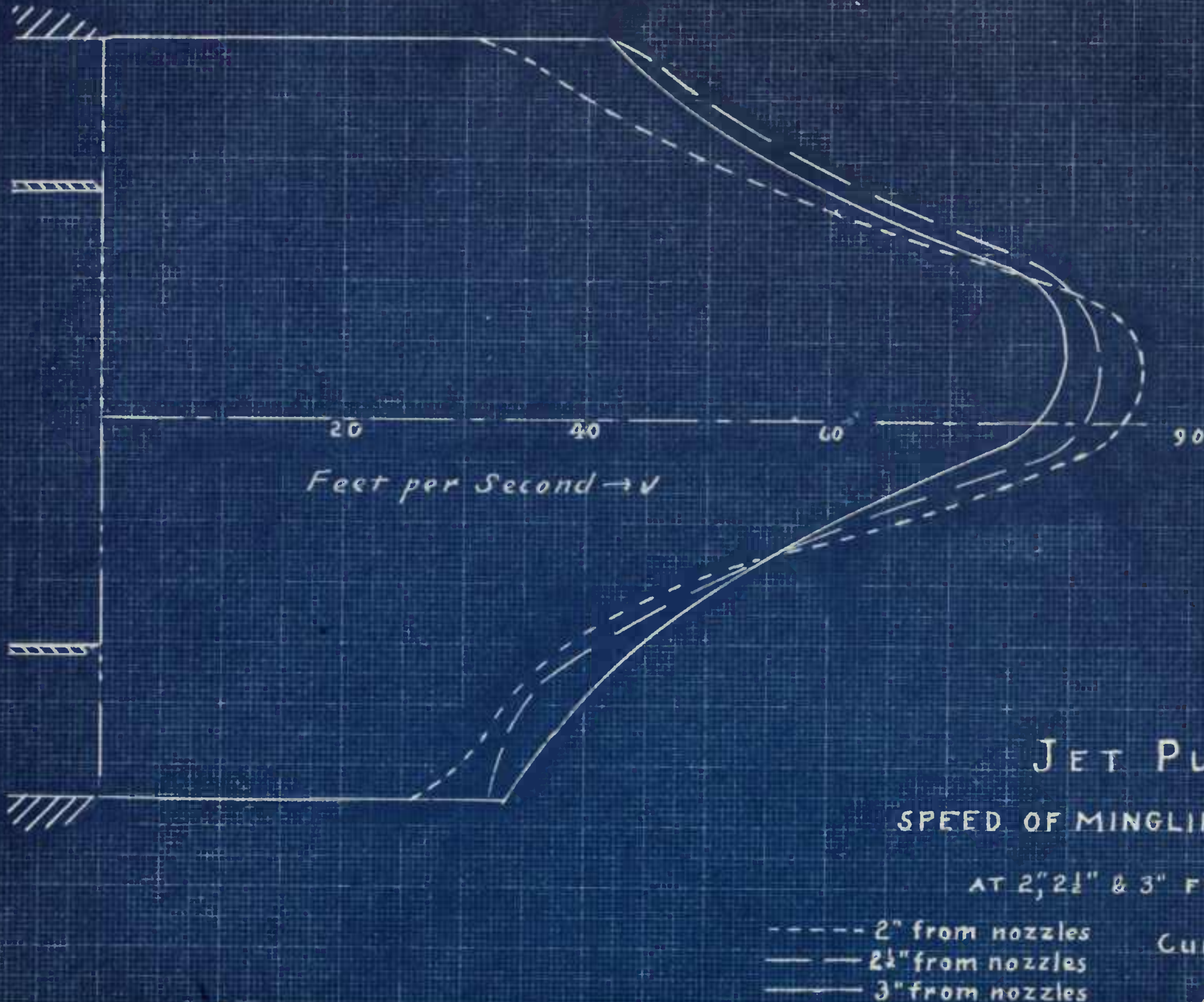
SPEED OF MINGLING STREAMS

AT 0", 1" & $1\frac{3}{8}$" FROM NOZZLES

- - - - 0" from nozzles
— — 1" from nozzles
—— $1\frac{3}{8}$" from nozzles

Curve Sheet No. 1

H. H. Bliss.



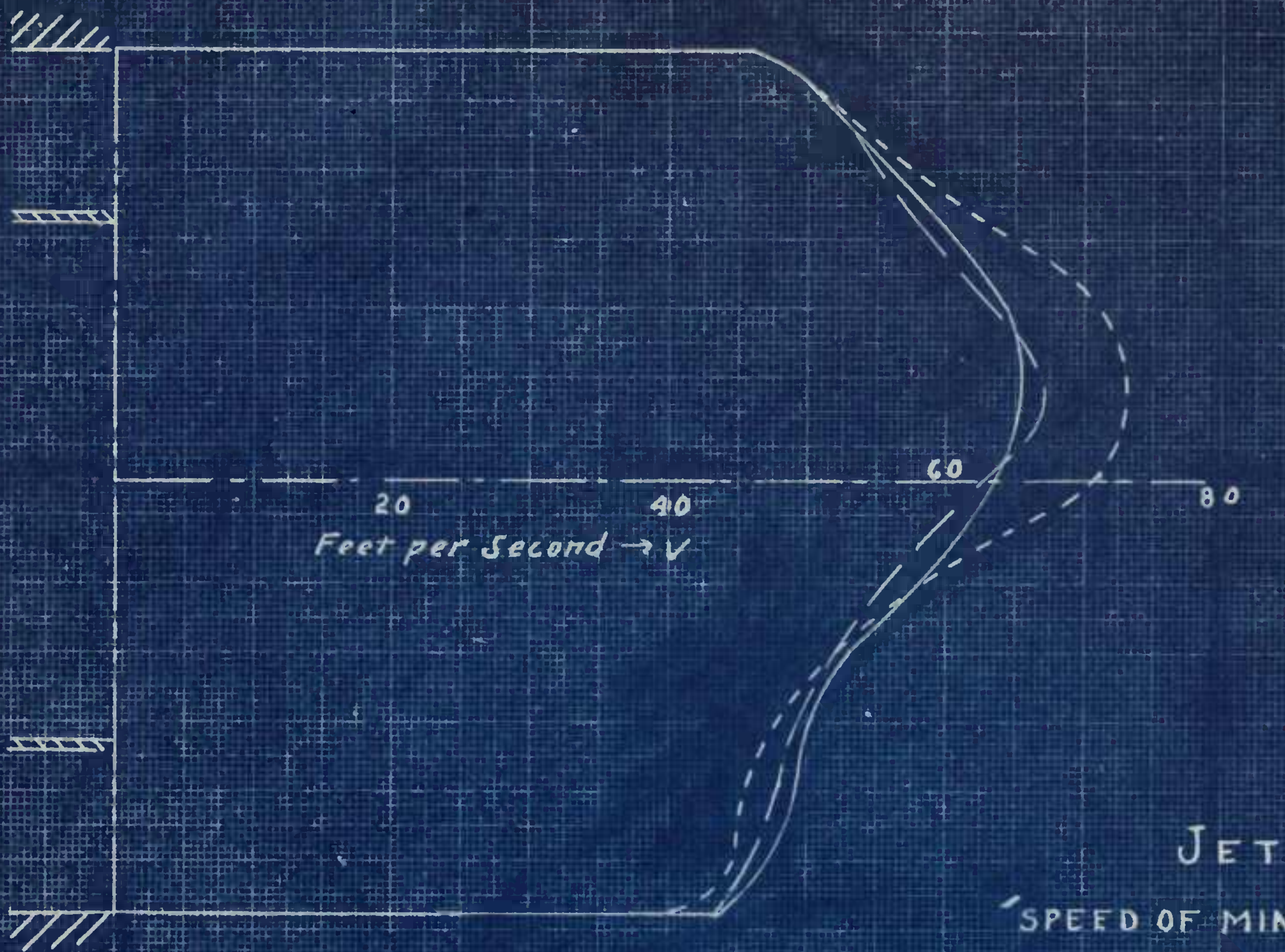
JET PUMP

SPEED OF MINGLING STREAMS

AT 2", 2½" & 3" FROM NOZZLES

Curve Sheet No. 2

H. H. Bliss



JET PUMP

SPEED OF MINGLING STREAMS

AT 4", 5" & 6" FROM NOZZLES

Curve Sheet No. 3

H. H. Bliss.

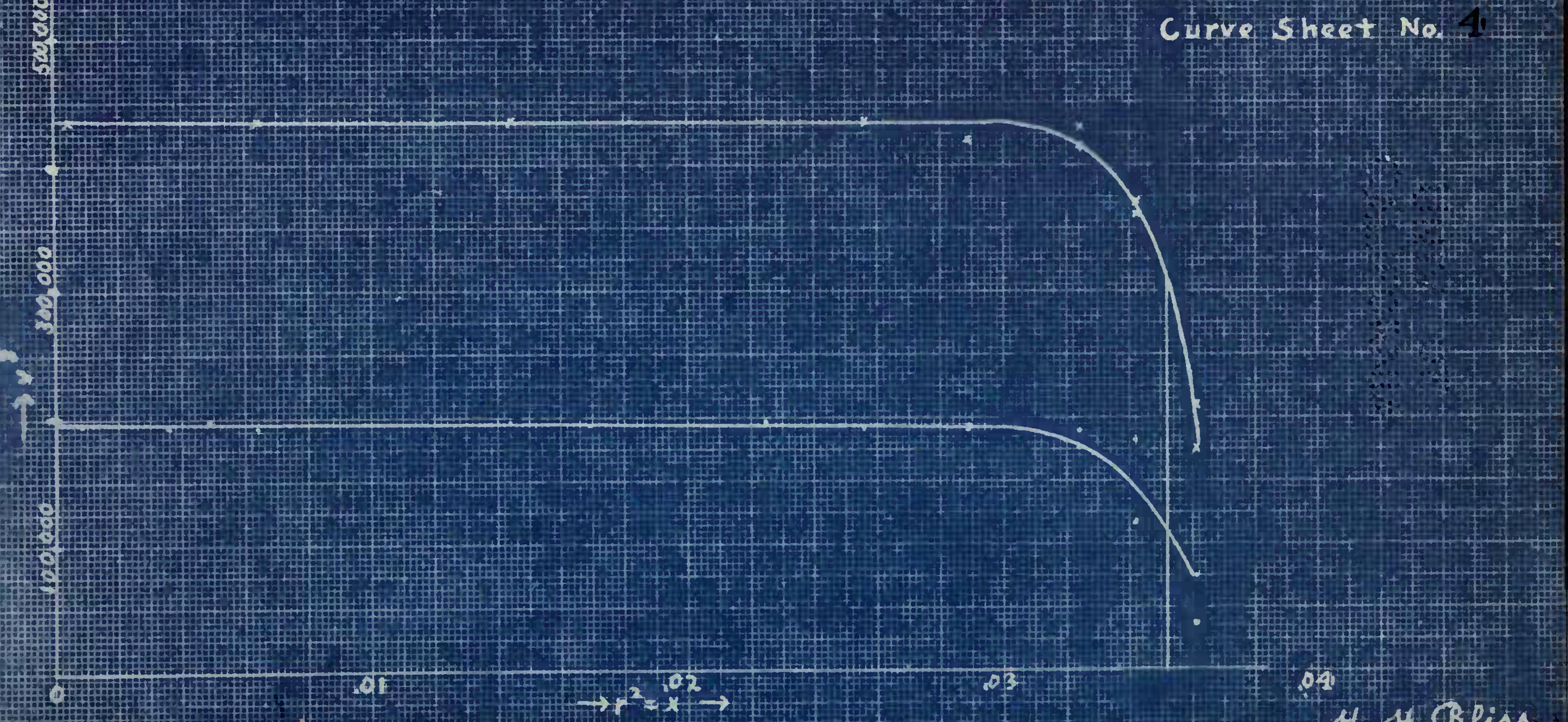
JET PUMP

INNER NOZZLE COEFFICIENT TEST

Upper Curve for Pressure = 39.6#□"

Lower Curve for Pressure = 23.0#□"

Curve Sheet No. 4



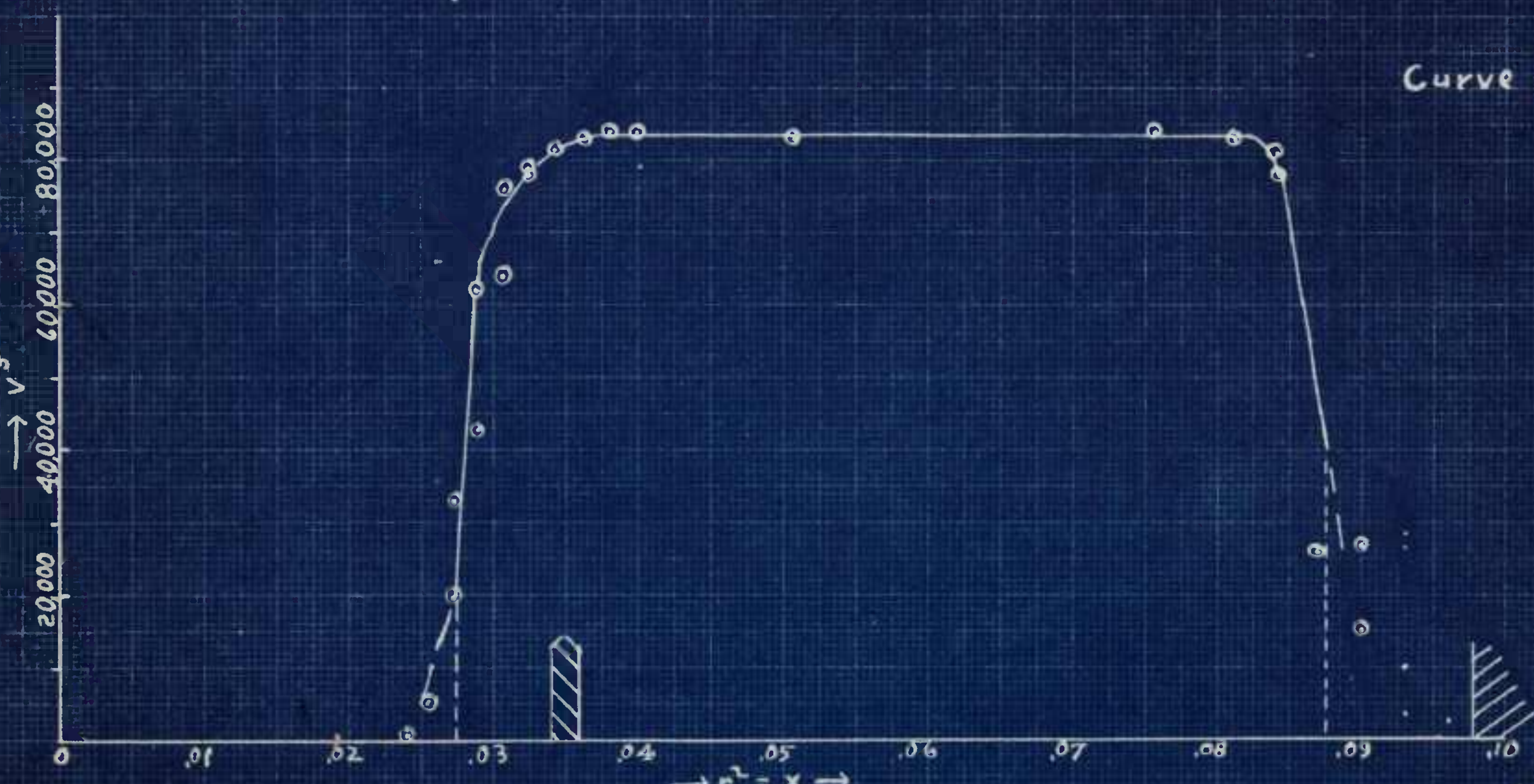
H. H. Bliss

Jet Pump

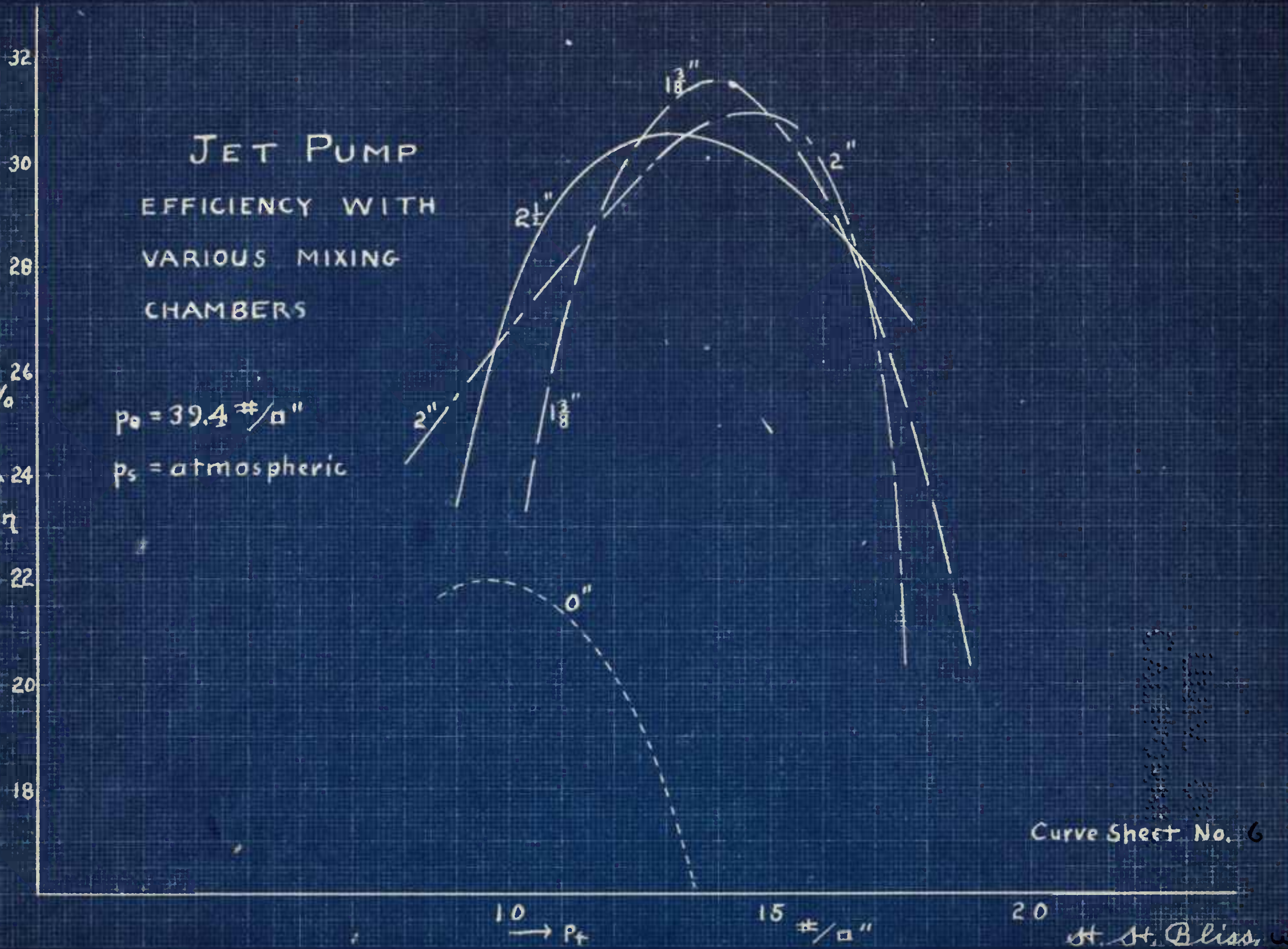
Outer Nozzle Coefficient Test

Pressure behind Nozzle = 12.9#a"

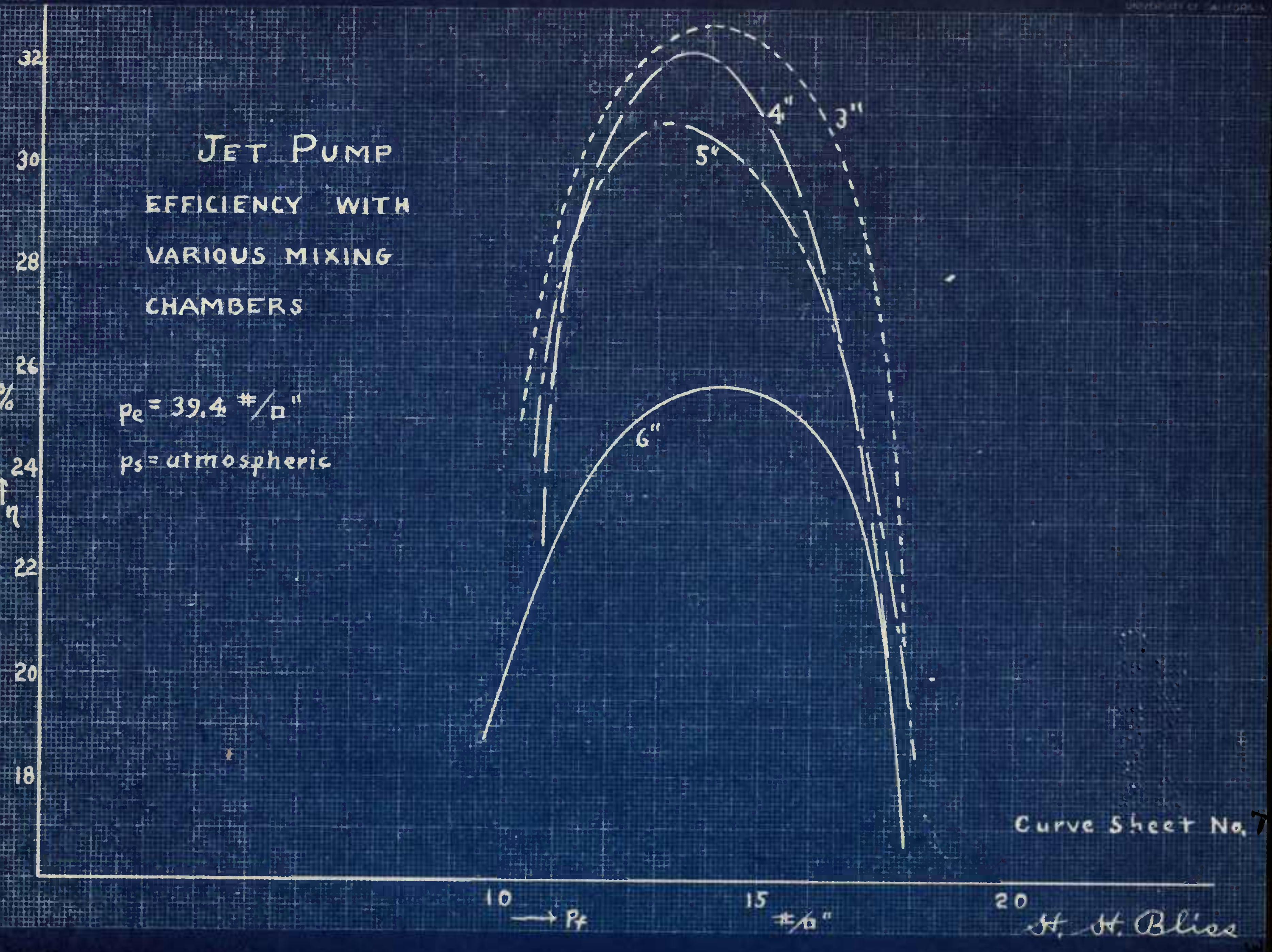
Curve Sheet No. 5



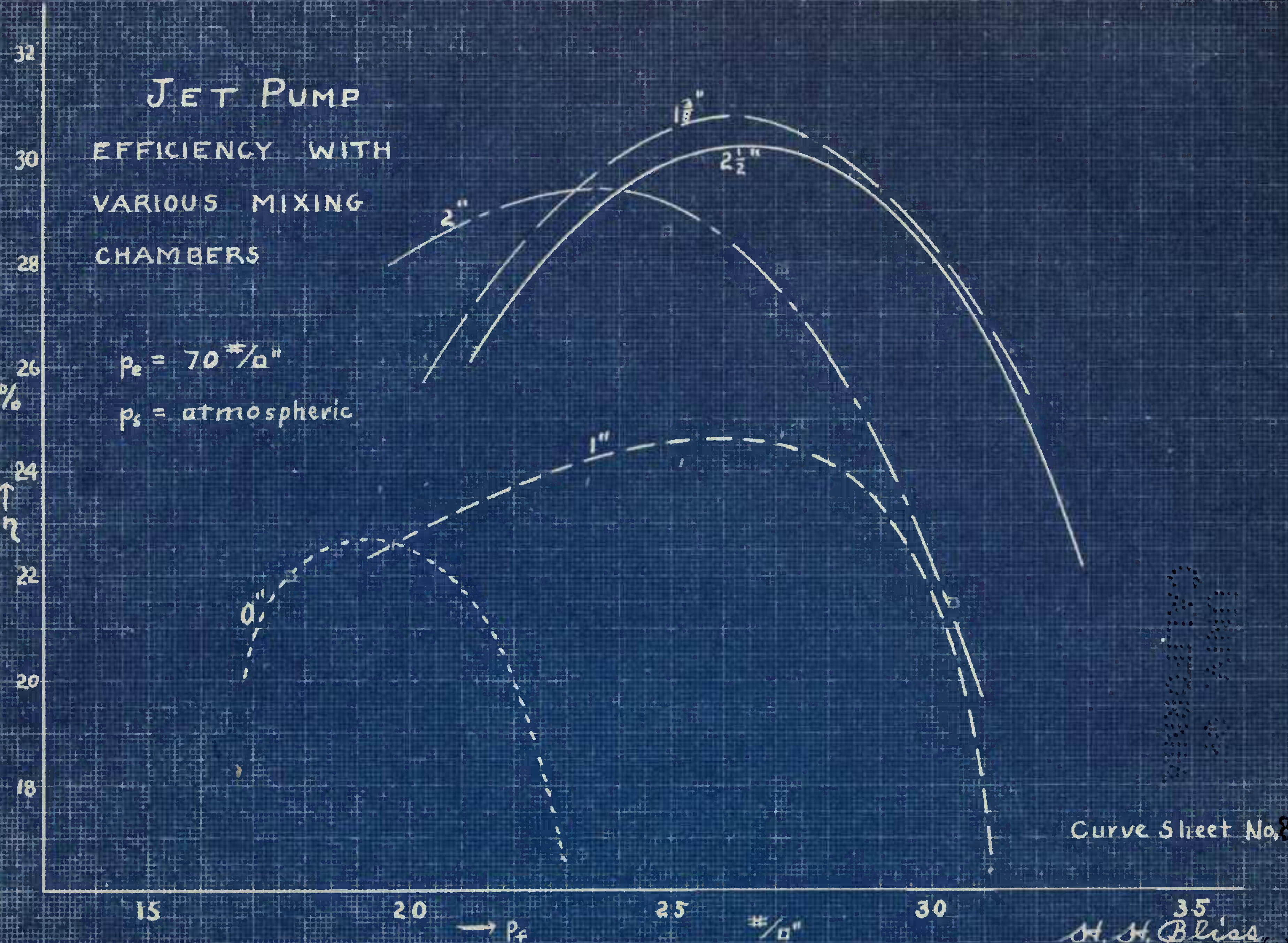
H. H. Bliss

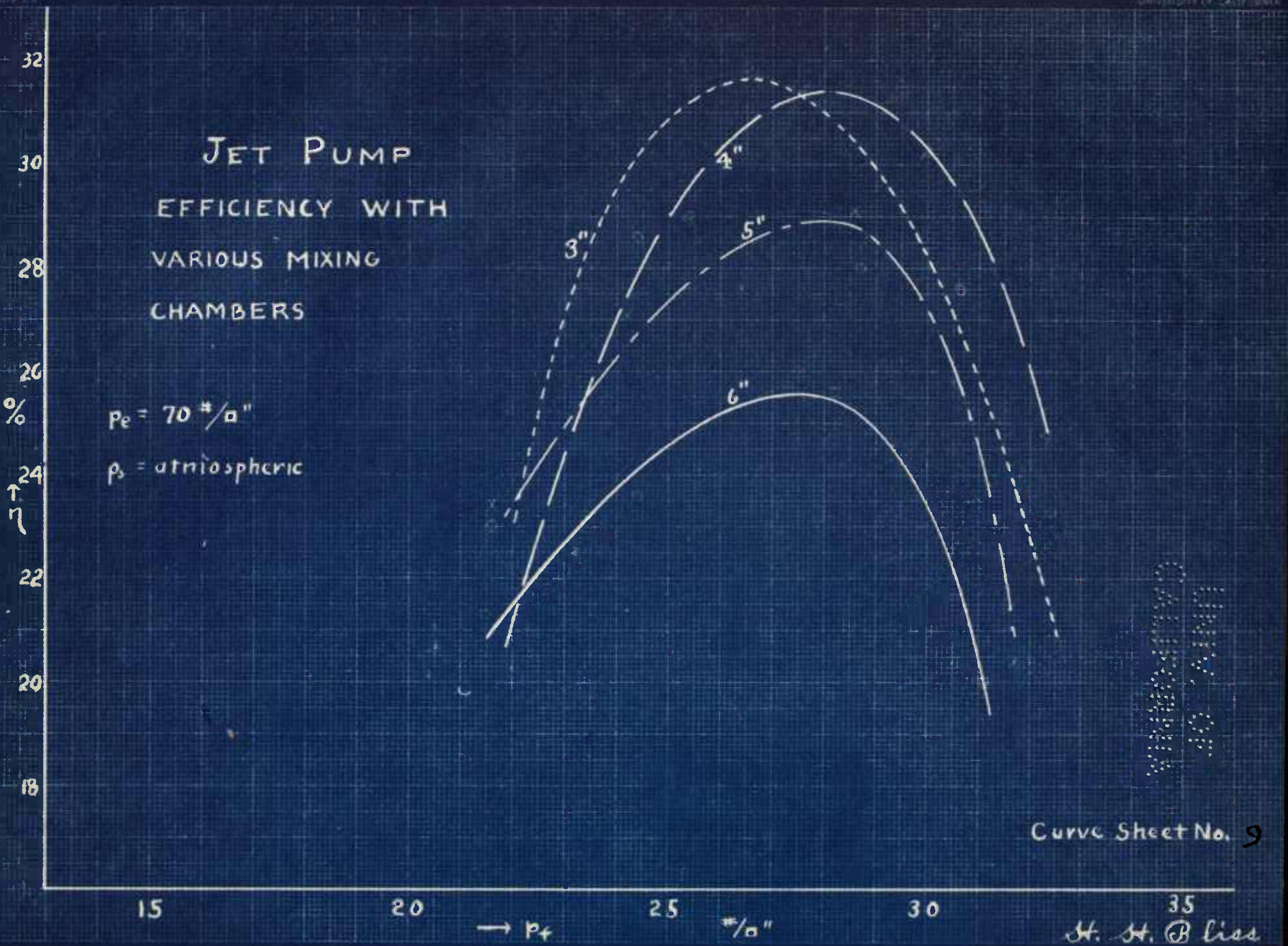


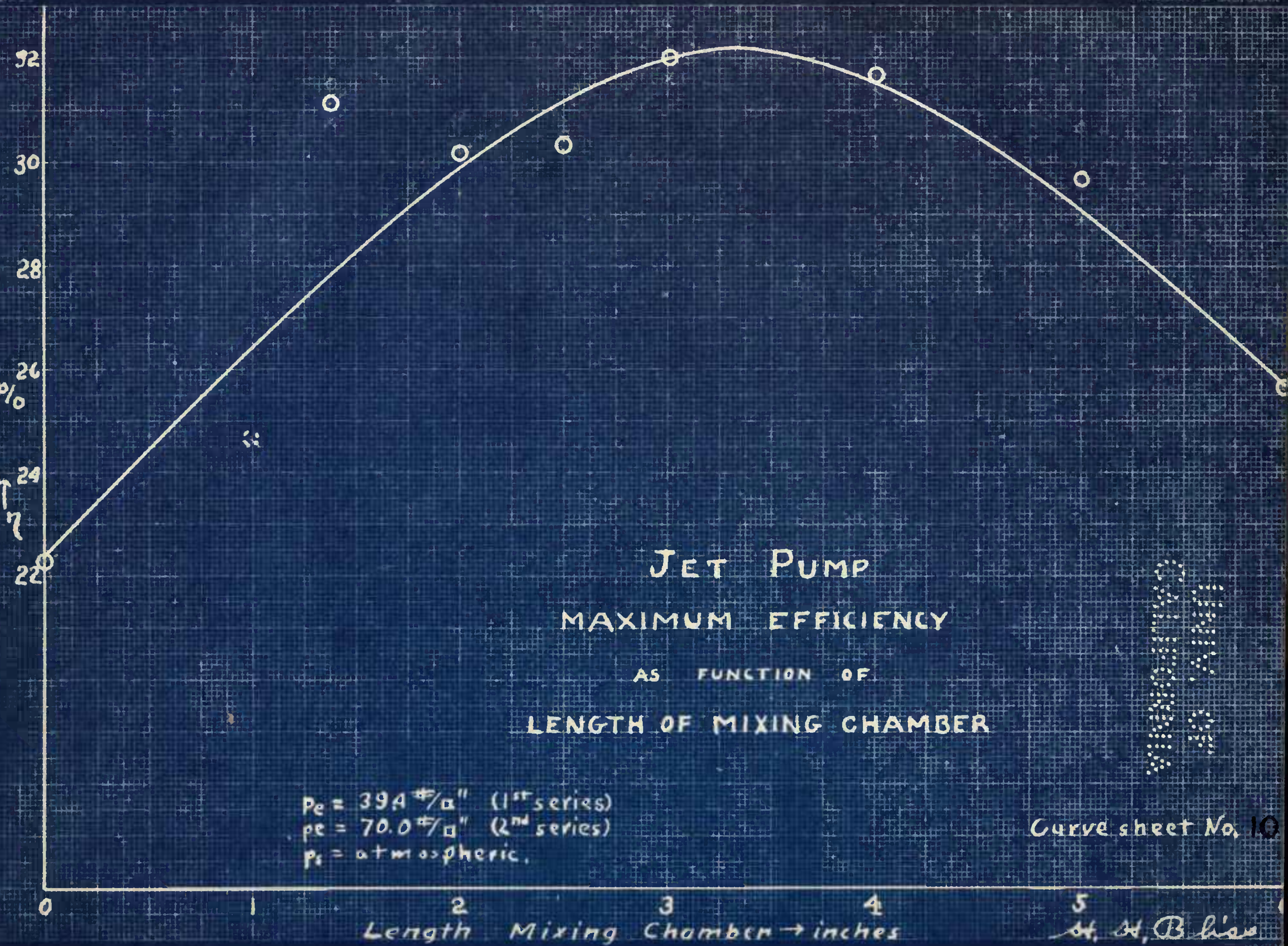
JET PUMP
EFFICIENCY WITH
VARIOUS MIXING
CHAMBERS
$p_o = 39.4$ #/□"
p_s = atmospheric
$1\frac{3}{8}$"
2"
$2\frac{1}{2}$"
2"
$1\frac{3}{8}$"
0"
32
30
28
26
%
24
η
22
20
18
10
15
20
→ p_t
#/□"
Curve Sheet No. 6
H. H. Bliss.



JET PUMP
EFFICIENCY WITH
VARIOUS MIXING
CHAMBERS
$p_e = 39.4$ #/□"
p_s = atmospheric
3"
4"
5"
6"
32
30
28
26
24
22
20
18
%
η
10
15
20
→ p_f
#/□"
Curve Sheet No. 7
H. H. Bliss


JET PUMP
EFFICIENCY WITH
VARIOUS MIXING
CHAMBERS
p_e = 70 #/□"
p_s = atmospheric
$1\frac{3}{8}$"
$2\frac{1}{2}$"
2"
1"
0"
32
30
28
26
24
22
20
18
%
↑ η
15
20
25
30
35
→ p_f
#/□"
Curve Sheet No. 8
H. H. Bliss


JET PUMP
EFFICIENCY WITH
VARIOUS MIXING
CHAMBERS
$p_e = 70$ #/□"
p_s = atmospheric
3"
4"
5"
6"
32
30
28
26
24
22
20
18
% ↑ η
15
20
25
30
35
→ p_t #/□"
Curve Sheet No. 9
H. H. Bliss



UNIVERSITY OF CALIFORNIA
JET PUMP
MAXIMUM EFFICIENCY
AS FUNCTION OF
LENGTH OF MIXING CHAMBER
Pe = 39.4 #/□" (1st series)
pe = 70.0 #/□" (2nd series)
ps = atmospheric.
Curve sheet No. 10
UNIV. OF CALIFORNIA
Length Mixing Chamber → inches
η %
0
1
2
3
4
5
22
24
26
28
30
32
H. H. Bliss